International Marine Engineering

OCTOBER, 1909.

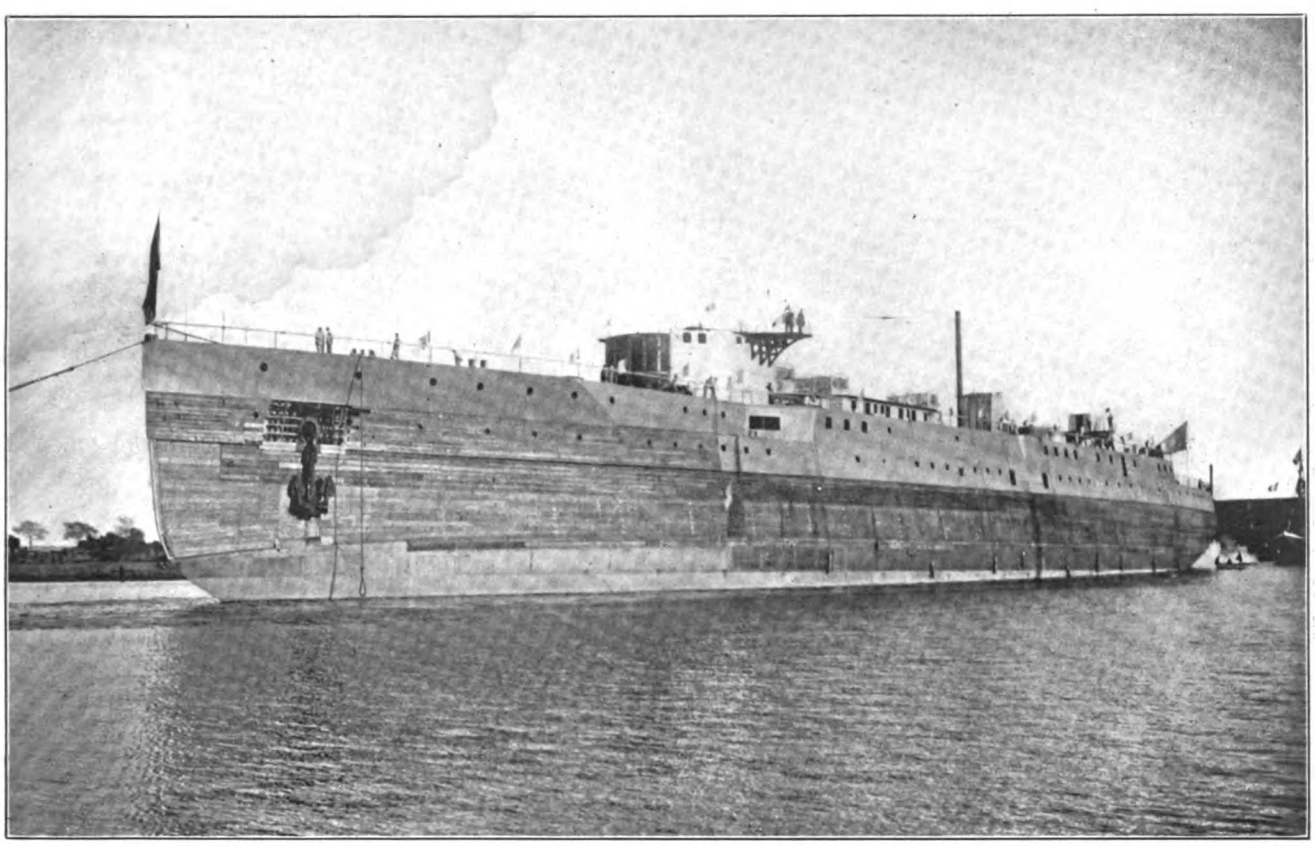
THE FRENCH BATTLESHIPS DIDEROT AND CONDORCET.

The *Diderot* and *Condorcet* are two of the six battleships of the *Danton* class authorized in the French naval programme of 1906-1907. The other ships of this class are the *Voltaire, Vergniaud* and *Mirabeau.* These ships are all of the same general dimensions and arrangement but differ somewhat in details. The first of these to be launched was the *Voltaire,* in January, 1909, at La Seyne, near Toulon. The *Diderot* and *Condorcet* were launched at St. Nazaire, the former on April 19, 1909, from the yards of the Chantiers de L'Atlantique, and the latter, April 20, from the Ateliers & Chantiers de La Loire. The other three ships are still on the ways, the *Verginaud* at the Ateliers de La Gironde, Bordeaux; the *Danton* at Brest, and the *Mirabeau* at Lorient. All, however, will soon be ready for launching.

The principal dimensions of the *Diderot* and *Condorcet* are as follows:

Length over all.....	481 feet.
Length on waterline.....	476 feet.
Extreme beam.....	84 feet 8 inches.
Depth, at full load, amidships.	27 feet 1 inch.
Full load displacement.....	18,235 tons.
Designed horsepower.....	22,500
Speed.....	19.25 knots.

These battleships have no metallic keels, simply a false keel and two docking keels of teak. The docking keels, as well as the steel bilge keels, extend for more than half the length of the ship amidships. The stem is of forged steel and the stern of cast steel. There are nine keelsons on each side of the main keel. Amidships, the frames are spaced about 34 inches center to center. The double bottom extends up to the lower protective deck. The outside plating varies from ⅜ to ¾ inch in thickness, or from 15 to 30 pounds weight. Behind the belt of side armor there is a double thickness of ⅜ inch or 15-pound plate. Numerous watertight compartments subdivide the hull from end to end. Many of these are not even pierced by watertight doors.



DIDEROT IN DOCK AFTER LAUNCHING, SHOWING EXTENT OF MAIN ARMOR BELT.

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The general protection of the ship is according to the French principle of the "caisson blinde" in connection with a "caisson cellulaire," protecting the ship as far as possible against torpedo attack. At the center line of the ship the lower protective deck is slightly above the load-waterline. At the sides it is 4 feet 10 inches below this, where it is connected with the upper edge of the "caisson cellulaire." An upper protective deck is worked at the height of the upper edge of the belt of side armor; that is, 8 feet above the load-waterline. This deck is constructed of three thicknesses of plate, each 5/8 inch thick. The lower protective deck is constructed of three plates each 5/8 inch thick on the flat, but on

STERN VIEW OF THE DIDEROT ON THE WAYS.

the slopes it is protected in addition by armor plates 4 inches in thickness.

As shown by the photographs, the main armor belt extends from the stem to within a few feet of the stern. It is composed of three strakes. The lower edge of the first strake is 3.15 inches thick amidships and at the sides of the 12-inch turrets, while the upper edge is 10.63 inches thick amidships and 7.87 inches at the sides of the 12-inch turrets. Forward and aft of this the lower edge of the first strake is 2.36 inches thick and the upper edge 7.09 inches thick, while at the ends the lower edge is 2 inches thick and the upper edge 3.15 inches thick.

The second strake is 9.84 inches thick amidships, 9.06 inches thick at the lower edge and 7.87 inches thick at the upper edge at the sides of the 12-inch turret; forward, it is 7.87 inches thick at the lower edge and 6.69 inches thick at the upper edge; while astern it is 5.41 inches thick, and at the ends 3.15 inches thick. The third strake is 2½ inches thick and about 115 feet long, extending from the stem to a point just aft the forward 12-inch turret. The armor belt is placed on a teak backing, which has an average thickness of 3.15 inches. Forward there is an athwartship armored bulkhead, 7.09 inches thick, extending from the sides to the 12-inch center line turret. There is a similar armored bulkhead, 7.8 inches thick, at the after 12-inch turret.

The 12-inch gun turrets are protected with 11.81 inches of steel armor and the barbettes with 11.02 inches of armor. The 9.4-inch guns are protected by 8.66 inches of steel in the turrets and 7.87 inches in the barbettes.

The space between the two protective decks and the armor belt, which is termed the "tranche cellulaire," is divided into numerous small compartments, with a passage at the center line of the ship. These compartments form bunkers and storerooms and give access to a cofferdam, which is worked throughout the entire length of the ship the total height of the armor belt. All passages, funnels, ventilators, etc., extending through the "tranche cellulaire" have been reinforced their entire height.

The "caisson cellulaire," which is designed to protect the ship as far as possible against torpedo attack, is constructed as follows:

About 8 feet 6 inches from the outside plating a vertical longitudinal bulkhead has been worked. This has an average height of 16 feet 6 inches and is made of plates having a total thickness of 1¾ inches. Behind the cofferdam and at a certain distance from it there is a second vertical bulkhead, .59 inch thick, extending from the double bottom plating to the lower protective deck. Forward and aft bunkers and other compartments provide the same protection as these bulkheads. This "caisson cellulaire" form of protection was used for the first time on board the *Czarwitch*, built in 1899 at the Forges & Chantiers de la Mediterrannée, and proved its efficiency by saving the ship from a total loss when under heavy torpedo attack.

Only one conning tower is fitted, and this is located on the navigating bridge, where a clear view can be obtained. It is noticeable that there are no obstructions on the bridge, as there are on all previous battleships or armored cruisers belonging to the French navy. The conning tower is protected by 11.81-inch armor, while the armored tube leading to it is protected by armor 8.66 inches thick above the upper protective deck, and 2.36 inches thick between the upper and lower protective decks.

From the foregoing it is evident that the protection of these ships is very good for an 18,000-ton battleship. Of course, it might have been better, but any additions would have necessitated a greater displacement, which, at the time these ships were designed, was impossible, owing to the condition of financial and political matters.

The main armament consists of four 12-inch guns mounted in pairs in two revolving turrets on the center line of the ship, one forward and one aft. There are twelve 9.4-inch guns, mounted in pairs in six revolving turrets, located on the spar deck. The secondary battery is composed of sixteen 3-inch quick-firing guns and ten 1.8-inch guns. It has been stated that two 18-inch submerged torpedo tubes would be fitted forward, but, up to the time of launching, no special arrangements were made for them, and it is now expected that the space and weight which they would occupy will be used for ammunition for the heavy guns.

The armament of these ships has been widely criticised, but the fact still remains that these boats have been built to fight in the Northern seas, where it is very likely that the future naval supremacy will be settled by the European powers. In these waters foggy weather makes it difficult to fight at a distance much over 3,000 yards. At this distance these vessels

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will be able to do effective work with their large quick-firing guns. At the same time such an armament is capable of making a good showing against any of the battleships belonging to the Triple Alliance in the Mediterranean. Of course, it must be admitted that this armament would be at a disadvantage in fine weather against the latest types of English or American battleships. It is practically certain that on future French battleships the *Dreadnought* idea will be carried out by using only big guns of a single caliber for the main battery, but in that case the displacement will be increased at least to 20,500 tons.

Just criticism can very well be made of the secondary battery of the *Diderot* and *Condorcet*. The 3-inch 12-pounder guns are not efficient against modern torpedo-boat destroyers. Four-inch quick-firing guns would at least be fairly efficient and it cannot be readily understood how such a small-caliber gun has been selected for this work. Also, the secondary battery is so arranged that it is practically unprotected, and the chances are that after an engagement this indispensable battery would be out of commission.

The forward and stern fire consists of two 12-inch and eight 9.4-inch guns. The broadside fire consists of four 12-inch and six 9.4-inch guns. The anti-torpedo-boat fire on each side consists of two 3-inch guns forward and four 3-inch guns in two casemates at the center of the ship, and two 3-inch guns aft, besides the 1.8-inch guns, which are located in various commanding positions on the bridge and spar decks.

It is very evident that in these ships the defensive qualities have been much better carried out than the offensive qualities. This of course is strange, since a battleship is built for fighting purposes and it is almost impossible to get the advantage over an enemy which has much heavier offensive powers.

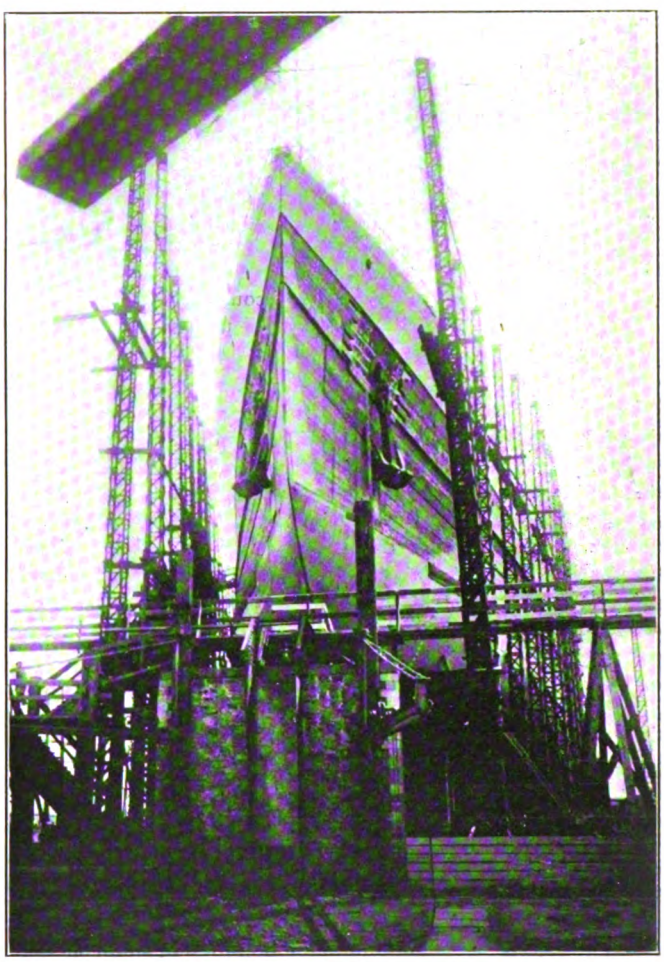
The ships will be propelled by four screws, each driven by Parsons turbine engines. There are eight turbines in each ship, four for full speed forward, two for the lower cruising speeds and two for astern speed. The dimensions of the high-speed turbines are as follows: High-pressure, diameter, 9 feet; length, 23 feet 10 inches. Low-pressure, diameter, 12 feet; length, 23 feet 6 inches. The high-pressure turbines drive the outside propellers, and the low-pressure turbines the inside screws. At full speed, the designed number of revolutions is 300 per minute. The high-pressure cruising turbines are on the inside shafts, the steam being exhausted into the high-pressure high-speed turbines and then into the low-pressure turbines. The cruising turbines are 8 feet 4 inches in diameter and 17 feet 6 inches long. The astern turbines drive the outside propellers and are 9 feet in diameter and 12 feet long. The inner shafts are located 7 feet from the center line of the ship, and the outer ones 21 feet 10 inches from the center line.

The *Diderot* and *Condorcet* will each be fitted with twenty-six Niclausse watertube boilers, each boiler having 1,560 square feet of grate surface and 14,150 square feet of heating surface. The normal steam pressure will be 257 pounds per square inch. During the forced-draft trials the pressure of water in the stokehold must not exceed 1.18 inches of water.

The boiler tubes are 3 5/16 inches outside diameter and the ordinary tubes 3 1/32 inches inside diameter. The reinforced tubes are 2 27/32 inches inside diameter. There are five funnels, 83 feet high above the grates, with maximum diameters of 8 feet 3 inches and minimum diameters of 4 feet 4 inches.

For the trials of these battleships the allowable coal consumption is calculated per mile and not per indicated horsepower hour. The contract figures were as follows: For the 10-hour full-speed trial, all boilers to be worked with forced draft, coal consumption per mile run, 2,060 pounds; per square meter of grate area, 287 pounds. For the 6-hour full-speed trial, with three-quarter boiler power and forced draft, the consumption per mile run is not stipulated, but the consumption per square meter of grate area is 397 pounds. For the 24-hour ordinary trial, with full boiler power at natural draft, the coal consumption is to be from 1,410 to 1,500 pounds per mile run and 183 pounds per square meter grate area. At low speed (10 knots), the consumption is to be from 573 to 618 pounds of coal per mile run. The total bunker capacity of each ship is 2,100 tons, and the approximate steaming radius at 10 knots speed, 8,130 miles.

The ships are lighted throughout by electricity and heated by steam. Only those auxiliaries requiring a large amount of power are driven by steam, all others are driven by electricity.



BOW VIEW OF THE CONDORCET JUST BEFORE LAUNCHING.

The dynamos are driven by turbine engines. Eight searchlights are to be fitted, two forward, two in the masts, two amidships and two aft.

The estimated cost of each of these ships is as follows:

Hull and machinery	$8,331,500	(£1,712,500)
Armament	1,939,566	(398,000)
Miscellaneous	35,967	(7,380)
Total	$10,307,033	(£2,117,880)

National Motor Boat Show

The 1910 National Motor Boat Show is to open at Madison Square Garden, New York City, at noon Feb. 19, and continue until Feb. 26. It has been decided to open the doors to the public every morning at 9 A. M., so as to afford business men and out-of-town parties an opportunity to visit the show at a time most convenient to them.

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SUPERHEATED STEAM IN MARINE WORK—III.

BY F. J. ROWAN.

STEAMERS FITTED WITH SUPERHEATERS AND RESULTS.

It is no bad evidence of the practical success of superheating steam to find that there are now afloat about 220 vessels of all sizes, from canal boats up to naval cruisers, which are equipped with superheaters. Of these Germany has the credit of having equipped 165 steamers (about 130 for the canals, lakes and rivers of the continent of Europe and for coasting service, some thirty sea-going steamers and four or five vessels of the Imperial navy); Britain about forty (all deep-sea steamers, including four cruisers of H. M. navy); America twenty (of which eight are naval vessels), and France six or seven (all merchant vessels). There may be some others, as it is not an easy matter to ascertain what is being done in such a matter, but we believe that our information is practically complete.

Taking the countries in the order named, it is to be observed that the steamers ascribed to Germany include the vessels belonging to that country and those owned in Switzerland, Italy, Austria-Hungary, Russia, Sweden and Holland, many of them having been fitted by engineers in those countries. They are, however, all fitted with one or the other of the various forms of the Schmidt superheaters (with one or two exceptions), and are worked on Herr Schmidt's system of high superheat. The first marine boilers fitted with these superheaters were built in 1898 by Messrs. Sulzer Bros., of Winterthur, and were supplied to steamboats plying on some of the lakes in Switzerland. The cost of coal being comparatively high in that country, the economy of fuel, due to the superheating, was a matter of great interest to the owners of such craft. Good results having been obtained from the Swiss lake steamers, the equipment of those running on the North Italian lakes, the Lake of Constance and on the rivers Oder, Danube, Volga, etc., soon followed. The horsepower of these lake steamers ranges from 100 up to 1,000, the majority being 300, 400 and 600 horsepower. The river steamers have from 130 to 800 horsepower, and following these came the larger Rhine steamers of 1,000 to 1,350 horsepower, with which "results of an extremely satisfactory nature were achieved."

Among the sea-going vessels fitted are the vessels of the Oldenbourg-Portugal Steamship Company, of Oldenbourg, and the Argo Steamship Company, of Bremen; these steamers having engines of 900 and 950 horsepower. The war ships include the cruisers *Dresden*, *Ersatz-Jagd* and *Ersatz-Schwalbe*, of 15,000 horsepower each, and the *Ulan*, a steam tender of 1,700 horsepower.

The total horsepower afloat working on the Schmidt system amounts to over 100,000, the superheaters being in nearly every case applied to the standard Scotch or cylindrical type of boiler. About forty of these boilers have the smoke-tube form (Fig. 16), and about ninety have the flame-tube form (Fig. 15); ten having smoke-box or funnel arrangements and three directly-fired superheaters and other special designs.

Regarding results, it is said that superheated steam of a temperature of about 660 degrees F. in compound marine engines gives a fuel economy of approximately 30 percent, as compared with the same type of engines using saturated steam, and about 15 percent as compared with triple-expansion engines also using saturated steam. Accurate records of the working of the Rhine steamers *F. Haniel I.* (of 1,000 horsepower) and *Hugo Stinnes I.* and *II.* (each of 1,350 horsepower), which have been running since 1905, with compound engines and superheated steam on this system, bear out the claim of economy. It has been found that in paddle steamers there is considerable loss of heat in the steam from the boiler, on account of the comparatively small number of revolutions per minute made by the engine. Consequently, with saturated steam the loss from condensation is greater than with stationary engines. Thus with superheated steam the saving in fuel (which is less than the saving in steam) is greater in paddle boats than with stationary engines of the same power, and amounts to about 30 percent with a superheat of 270 degrees F. On screw steamers the fuel economy amounts to about 25 percent for the same degree of superheat.

In the case of twelve steamers belonging to the two steamship companies mentioned above, which have triple-expansion engines and Schmidt smoke-tube superheaters fitted in their existing boilers, they have realized in continued service an economy of 18 to 20 percent in coal as compared with a saturated steam plant of the same size. The steam pressure in these ships is 185 pounds per square inch, and the average temperature of the superheated steam is 608-644 degrees F. The original piston valves on the high-pressure cylinders were retained and work satisfactorily. In the case of engines having slide valves, however, it is found that a superheated steam temperature not exceeding 518 degrees F. must be used. Efficient lubrication is of great importance in both arrangements, and the method adopted is that of forcing the oil by a pump for the slide valves, or by a lubricator, which forces it into the center of the main steam pipe for the piston valves. These steamships indicate on the average 1,000 horsepower; have a boiler-heating surface of 3,220 to 3,440 square feet, and a superheating surface of 1,180 to 1,290 square feet. Where forced draft on Howden's system is used average economies of 25 to 30 percent in coal have been realized.

In Britain, the first superheating installation made by the Central Marine Engine Works, of West Hartlepool, was in the year 1891. This apparatus, then designated a "steam drier," was fitted on board the steamship *Elmeto*, owned by the London & Northern Steamship Company, and consisted of a series of oblong, cast iron hollow bulbs, bolted together and communicating with one another—these occupying the whole space of the smoke-box immediately above the top row of boiler tubes. The steam passed through the interior of these bulbs, the heated waste gases escaping from the boiler tubes coming in contact with their exterior surface, and this apparatus, though of a simple character, was sufficient to demonstrate that economy could be secured by using partly dried steam, and that the waste gases from boilers could be beneficially used for this purpose.

A few years afterwards the same engineering works fitted in the steamship *Inchmona* a complete set of apparatus designed for superheated steam, including boilers for a pressure of 255 pounds per square inch, superheaters (Fig. 9), Ellis & Eaves' system of induced draft and quadruple-expansion engines on five cranks. The results obtained were sufficiently encouraging to induce the owners to fit up succeeding steamers with the same system, the *Inchdune*, *Inchmarlo* and *Nassovia* being the first of the fleet, which also includes the *Inchkeith*, *Incharan* and *Inchmore*. The two first named were placed in commission in 1900, and were succeeded by the other vessels. "These are fine freight vessels of 6,000 tons, fitted with quadruple-expansion, five-crank engines, one high-pressure, two intermediate and two low-pressure cylinders; air, bilge and feed pumps being driven off the main engines. The working pressure at the boiler is 267 pounds per square inch, and the waved-tube superheaters are set in the up-takes, just above the upper row of tubes of the two Scotch boilers, each of which is 13 feet in diameter by 10 feet 6 inches long. The steam is superheated from 412 degrees F. (the temperature due to the pressure) to 469.5 degrees, or a superheat of 57½ degrees F. The temperature maintained in the second receiver and at the high-pressure steam chest averaged 447 degrees. The waste gases, in addition, heat the air for combustion in the furnace about 250 degrees F., or from an out-

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side temperature of 54 degrees to 299 degrees F., the draft being controlled by the Ellis & Eaves' apparatus. The waste gases below the superheater have been found to be at 620 degrees, reduced by the superheater to 543 degrees and by the air heater to 404 degrees F., this temperature of chimney discharge showing over 33 percent of the waste heat turned into useful work."

On a trial of the *Inchdune* and *Inchmona*, between West Hartlepool and Dover, the coal consumption is reported to have been only 0.97 pound per indicated horsepower-hour, and

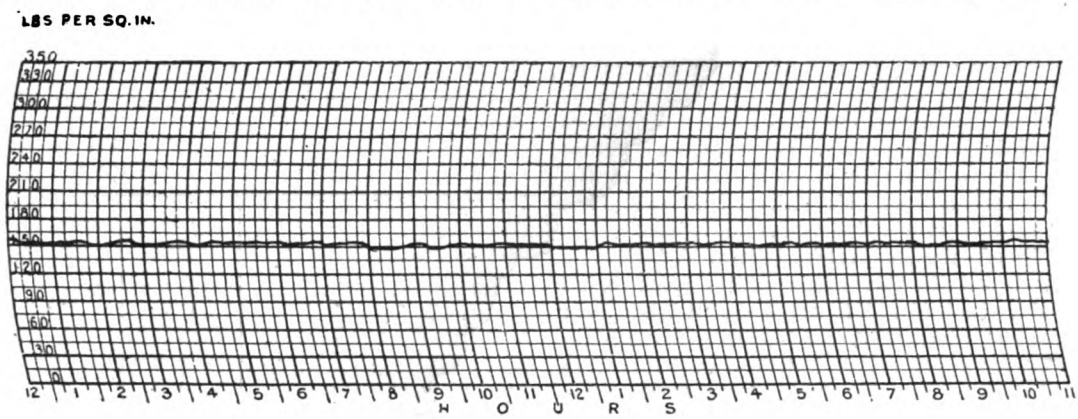

FIG. 20.

the *Inchdune* and *Inchmarlo* have been able to complete their voyages on an average consumption of 1 pound of coal per indicated horsepower-hour. The *Inchkeith* has for her twelve months' consumption in voyages between New York and Bombay an average rate of 1.1 pounds per indicated horsepower per hour. The total degree of superheat imparted to the steam was in the later cases 70 degrees F., making the temperature at the high-pressure steam chest 460 degrees F., and no ill effects have been noticed in the working of the engines, due to this temperature. It was stated by Mr. W. Bloor that "the diagrams showed an area of 89.7 percent of the area between highest and lowest pressure, and that comparative diagrams with saturated steam showed superiority in area of the superheated vapors as high as 15 percent.

Messrs. Thos. Wilson, Sons & Company, Ltd., at a comparatively early date. The first steamer so fitted was the *Claro*, of 5,350 tons displacement, and from 9 to 9½ knots speed, and her first voyage was made in November, 1900. That vessel was fitted with forced draft, a superheater in the funnel, and an air heater. The funnel gases were discharged at a temperature of about 420 degrees F., the temperature of the steam being from 490 degrees to 520 degrees F. at the engine stop valve, the temperature depending to some extent on the quality of the coal. The machinery has run, with the exception of some few initial troubles, without any bother whatever. The troubles at the commencement were due entirely to ignorance of the requirements imposed by the dryness of the steam at the above temperature; but having overcome these, no more trouble has been experienced than with ordinary machinery. The steam pressure was nominally 200 pounds per square inch, the engineer usually working with about 195 pounds on the boiler and 190 pounds on the engine, the superheat being thus about 120 degrees F. average in the high-pressure cylinder. The vessel having given very satisfactory results, it was decided to fit a superheater in the *Colorado*, a vessel of 8,400 tons displacement and 11½ knots speed, employed in the Atlantic trade. This was done in 1902, and the vessel has been running with the superheater since March of

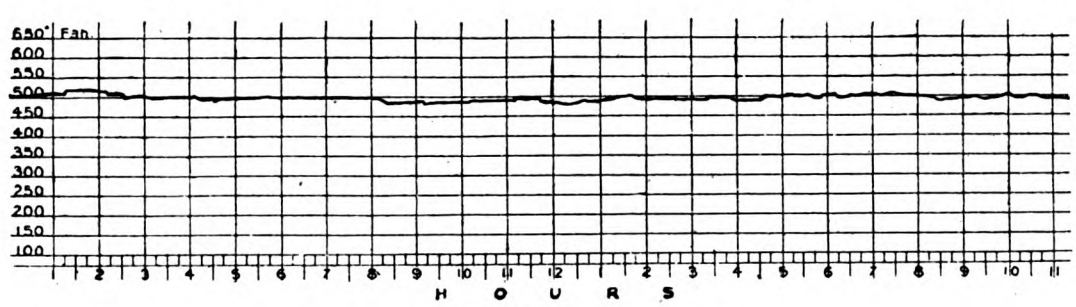

FIG. 21.

At a speed of 9½ knots, a consumption of 1.1 pounds works out to only 17 tons per diem, and at an average price of $4 (16/8) per ton the cost of fuel would be 22 cents (10d.) per mile, or at 6,170 tons dead weight 1 ton is carried 280 miles for a cent (½d.). The Central Marine Engine Works have also designed and fitted ships with other arrangements of superheaters to suit altered conditions with good results. One of these vessels had about 3,000 horsepower with four-crank, quadruple-expansion engines, Howden's forced draft arrangements and superheaters, but was, unfortunately, sunk in collision with an iceberg at an early period of her career.

Mr. W. S. Hide is one of the pioneers in this field, having commenced to introduce superheaters into the vessels of that year. The boilers are worked with natural draft, and a temperature of 500 degrees F. at the engine stop valve is attained, the steam pressure being 160 pounds per square inch.

Figs. 20 and 21 give records, over twenty-four hours, of the recording thermometer fixed for a voyage to the engine stop valve, and also of the recording pressure gage, and these show how steady both steam pressure and temperature are kept in practical working, with fires cleaned in the usual course.

The *Aleppo*, a vessel engaged in the Indian trade, of 8,600 tons displacement and 9 knots speed, was next fitted, giving similar results to the above, the boilers being worked with natural draft. The *Martello*, a vessel similar to the *Colorado*,

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fitted with watertube boilers of the Babcock & Wilcox type, under natural draft, had a superheater in the funnel, her steam pressure being 210 pounds. The *Idaho*, a vessel of 11,100 tons displacement and 11½ knots speed, with engines of the quadruple-expansion type, working at a pressure of 215 pounds per square inch, and boilers with forced draft, has a superheater and an air heater. Her steam temperature at the engine stop valve is from 510 degrees to 520 degrees F., and the funnel temperature from 350 degrees to 360 degrees F., and the consumption of coal is quite satisfactory.

No deterioration has been observed in the superheater tubes of the *Claro* after upwards of five years' running. Altogether, there are now fifteen vessels of this fleet fitted with superheaters, all of which are working satisfactorily. The superheater is placed at the base of the funnel, and thus utilizes the waste heat of the escaping gases. The steam temperature at the engine stop valve ranges from 500 degrees to 600 degrees F. in the various ships. No noticeable advantage has been found by working at a higher temperature, but there is a distinct loss directly the temperature falls much below 500 degrees F. By taking the temperature of the steam in an intermediate valve chest, Mr. Hide found that it takes from 80 to 100 degrees of superheat to do the work in the high-pressure cylinder, any superheat above that amount appearing in the intermediate-pressure valve chest. A superheat of less than 80 degrees has not been found to be of any practical value or to show any decided economy; above that degree a gain of 12 to 16 percent may be counted upon, depending on the various contributing factors, such as type and design of engine, amount of superheat, pressure of steam, etc. Difficulties are occasionally experienced with piston rings and glands in marine engines, which are absent from practice on land; and Mr. Hide accounts for this by the fact that the land engine is always maintained steady in a vertical position, whereas, this is not always so on board ship. The difference in the running of the engines with superheated and with saturated steam is very marked; directly the superheated steam is turned on all leakages of water at the glands are stopped, and the engine runs quite dry. The oil used for lubrication is a specially pure hydrocarbon oil, having a flash point of about 700 degrees F., and is quite satisfactory; the amount used in an engine of about 1,800 indicated horsepower being about 2 quarts per twenty-four hours, including the swabbing of all rods.

One of the Wilson fleet is the steamer *Martello*, which is fitted with Babcock-Wilcox boilers; and although fitted with four boilers she is able to maintain full speed on her voyages with three boilers, having 8,055 square feet of heating surface and a superheater surface of 2,000 square feet, the degree of superheat being 90 degrees F. Generally, in the working of these vessels the coal consumption per day, or per voyage, is found to compare very favorably with that of other similar ships.

Some of the steamers of the Great Eastern Railway Company's fleet were fitted some years ago with Watkinson superheaters (Figs. 11, 12). Of these, the twin-screw steamship *Yarmouth* was built by Messrs. Gourlay Bros. & Company, Dundee; the steamships *Clacton* and *Amsterdam* by Messrs. Earle's Shipbuilding Company, Ltd., Hull; the superheaters being made by Messrs. Meehan & Sons, Ltd., Glasgow. Beyond the fact that on trial a saving of about 18 percent was realized in fuel as compared with a sister ship with saturated steam, there are no data available about the performance of these vessels. The Allan Line steamship *Mongolian* was fitted with a Watkinson superheater of the independently-fired type, and although it worked well the engines gave trouble by frequent breaking of the piston rings. This was afterwards found to be due to friction caused by a deposit from the decomposition of the oil by heat.

H. M. S. *Britannia* is a first-class battleship of 18,000 indicated horsepower, having eighteen Babcock-Wilcox boilers, with a total heating surface of 40,020 square feet and a total grate area of 1,250 square feet; and three cylindrical boilers, with a total heating surface of 8,100 square feet and grate area of 247 square feet. Six of the Babcock-Wilcox boilers are fitted with Babcock-Wilcox superheaters (Fig. 14), the tube-heating surface of these six boilers being reduced by the amount of superheating surface. The following table gives the results of the official trials. To compare with the thirty hours' trial of six boilers with superheaters, a run of the same duration was made without superheaters, the engine and other conditions of working remaining the same; and in this trial the advantage with superheated steam was a gain of 15 percent.

RESULTS OF OFFICIAL TRIALS.

	Low Power.	Maximum Continuous.	Full Power.
Date of trial	July 9 & 10, 1906	July 14 & 15, 1906	July 17, 1906
Duration of trial	30 hours.	30 hours.	8 hours.
Number of boilers in use	6 B. & W. with superheaters	18 B. & W. & 3 cylindrical.	18 B. & W. & 3 cylindrical.
Heating surface (incl. superh.), sq. ft.	13,308	48,120	48,120
Grate area, square feet	413.4	1,497	1,497
Fuel, kind used	Welsh coal	Welsh coal.	Welsh coal.
Steam, average gage press., lbs. p. sq. in.	200	199	198
Draft, ins. water pressure in stokehold	.57	.325	.87
Indicated horsepower	3,410	13,078	18,624
I. H. P. per square foot of grate	8.25	8.73	12.44
Coal, total consumed per hour, lbs.	6,036	19,617	34,082
Coal per I. H. P. per hour, lbs.	1.77	1.50	1.83
Coal per sq. ft. firebox surface per h. lbs.	14.60	13.10	22.76
Heating surface per I. H. P.	3.90	3.68	2.58
Revolutions per minute	73.7	113.5	127.3
Uptake temperatures	347°F.	348°F.	423°F.
Steam temperature at boilers	200°F.	199.4°F.	198°F.
Degrees Fah. superheat at boilers	92.5°F.	72°F.	87.5°F.
Degrees Fah. superheat at engines	83°F.	33°F.	31°F
Feed temperature	65.6°F	80°F.	103°F
Total water per I. H. P., lbs.	18.19	16.21	18.55
Water per lb., coal, lbs.	10.276	10.8	10.136
Water per sq. ft. heating surface, lbs.	5.17	4.5	7.37

H. M. S. *Medusa*, a small cruiser, was fitted with eight Dürr boilers, having superheater tubes for trials at the instance of the committee on naval boilers in 1904. The total area of surface in the generating tubes was 21,369 square feet, and that of the superheater tubes was only 1,119 square feet, the tubes having an external diameter of 2¾ inches. So small a superheater could act practically only as a steam drier, but fairly good results were obtained on trial, and H. M. S. *Roxburgh* was fitted out with the same boilers and superheaters (Fig. 13), some details being improved. Seventeen boilers, with a collective heating surface of 41,600 square feet; a superheating surface of 2,245 square feet; a grate area of 1,085 square feet, and a working pressure of 220 pounds per square inch, supply steam for 16,000 indicated horsepower. No comparative results or trials with these boilers are published, but when they were tested on shore they were found to give extremely dry steam.

It is expected that H. M. S. *Bristol*, which is being fitted with Curtis turbines, will also have superheated steam, as that type of turbine is favorably affected by steam of that quality.

In America, the first trials with modern superheaters in marine practice were carried out in the steamer *J. G. Wallace*, fitted with Babcock-Wilcox boilers of 5,800 square feet of heating surface and 830 square feet of superheating surface. The superheaters were placed in the first pass of the hot gases, but were arranged so that they might be cut out of circuit by an alteration of the baffling, and thus the same boilers could be used for trial in the same ship with and without superheaters. The result was that an increased coal consumption of 16½ percent appeared to be due to the use of saturated steam.

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Two sister ships of the *J. G. Wallace* were fitted with superheaters, and also following these trials the American navy department fitted superheaters to four of the eight boilers of the *Indiana*. The results of her trials have not been published; but they must have been favorable, from the fact that other warships were subsequently ordered to be fitted with superheaters. In naval vessels the combination of marine turbines with superheated steam is a matter of great engineering interest.

The Southern Pacific Railroad Company's twin-screw steamer *Creole* was fitted with Curtis turbines and ten Babcock-Wilcox boilers with superheaters, using natural draft. The heating surface in the ten boilers is 28,500 square feet, superheater surface 4,350 square feet, grate surface 770 square feet, and the working steam pressure is 250 pounds per square inch. The turbines are of 8,000 horsepower aggregate. The thermodynamic efficiency (according to *Brassey's Naval Annual*, 1908), with steam 250 pounds pressure, works out at 44 percent at 250 revolutions per minute, whereas the *Dreadnought*, with Parsons turbines, showed a thermodynamic efficiency of 50 percent at about 250 revolutions per minute and 61.5 percent at about 330 revolutions. The *Creole's* steam consumption was fully 16 pounds per equivalent indicated horsepower per hour, with steam superheated 74.5 degrees F. It is said that some trouble was experienced with the turbines in later working; and as this was the first example of the application of Curtis turbines to marine work, that is not to be wondered at, if it is a true report.

The United States Navy Department made comparative tests on one of the lake steamers equipped with quadruple-expansion engines, and found that a net coal saving of 14.8 percent was effected by superheating the steam about 80 degrees F. This and previous experience with superheating induced the Mallory Steamship Company to equip their twin-screw steamer *Brazos* with Foster superheaters (Fig. 19). This vessel has quadruple-expansion, four-crank engines of 7,000 horsepower, and eight single-ended Scotch boilers, 14 feet diameter and 11 feet 9 inches long, with 18,000 square feet of heating surface, and a total grate surface of 472 square feet. One Foster superheater serves for each group of four boilers, the total superheating surface being about 4,000 square feet. They were each designed to add 65 degrees F. superheat to 45,000 pounds of steam at 215 pounds pressure per square inch, giving the steam a final temperature of 460 degrees F. The boilers are equipped with the Howden arrangements for forced draft, and the temperature of the gases passing from the superheaters to the air heaters is from 650 to 700 degrees F. The consumption of coal as fired has not exceeded 1.25 pounds per indicated horsepower-hour, and a speed of 14.5 knots has been maintained on 67 tons of coal per twenty-four-hour day.

In France, the Compagnie Générale Transatlantique fitted their steamer *La Rance*, in 1906, with vertical, triple-expansion engines, having Lentz valve gear with poppet valves, cylindrical boilers and Pielock superheaters (Fig. 18). The total heating surface of the boilers was 3,767.5 square feet, and the superheating surface 785.8 square feet. The steam pressure was 177 pounds per square inch and the temperature 518 degrees F., or about 140 degrees F. of superheat. Comparing *La Rance* with a sister ship, the *Garonne*, with similar engine, but with slide valves, and with similar boilers, without superheaters, trials showed the advantages due to superheat to be an increase in power of 18.1 percent and a decrease in coal consumption of 20.1 percent in favor of *La Rance*. Following these good results the same company introduced the Pielock superheater into the steamships *Perou*, the *Caroline* and the *Honduras*, and others are likely to follow. The *Perou* has been also compared in running with a sister ship, the *Guadeloupe*, identical but for superheaters and valve gear, and has shown a gain of 0.35 knot in favor of the *Perou*. Continuous diagrams of steam temperature showed that with superheaters the temperature variations do not exceed 36 degrees F. from the time of starting the engines to that of running at full power, including the period of cleaning fires.

The conditions of the successful employment of superheaters in marine practice may now be said to have been well established, and we may expect in the future a considerable development of the practice, especially in conjunction with turbines, as it is claimed that the steam consumption of the turbine may be reduced 1 percent for each 10 degrees F. of superheat.

REMODELING GASOLINE (PETROL) ENGINES FOR PRODUCER GAS.*

BY H. F. SMITH.

Among manufacturers of internal combustion engines who have, up to the present time, been confining themselves largely to gasoline (petrol), there is a strong temptation to enter the producer gas field, not by redesigning an entirely new engine but by undertaking to adapt the existing design to the requirements of producer-gas work. In many instances this can be done with a fair degree of success, but the degree of success obtained will depend entirely on how well the essential conditions of design requisite for operation on producer gas can be embodied in the existing structure.

There has been in the past an unfortunate impression among builders of this class of engines that about all that is required to make a gasoline (petrol) engine successful for producer gas is to increase the compression to about 150 pounds per square inch. As a matter of fact this is perhaps one of the least important of the several requirements for successful operation on producer gas. There is no reason why compression with producer gas is a matter of any greater importance than it is with gasoline (petrol). A gasoline (petrol) engine, operating under an initial compression of 80 pounds per square inch, will give a higher efficiency than one operating under an initial compression of 30 pounds per square inch. The same may be said of engines for producer gas with exactly the same force, except that with producer gas it is practical on account of the characteristics of the fuel to carry the compression to points that cannot be tolerated for gasoline (petrol). Experience has shown, however, that under ordinary conditions the heat losses introduced by abnormally high compression, together with the increased friction and strain on the various parts of the engine soon become greater than the increased efficiency of the cycle, and there is, therefore, very little gain, and, in some cases, a positive loss, by carrying compression above comparatively moderate limits. No further consideration need be given this point than to note that 150 pounds per square inch gage is perhaps more ordinarily used than any other pressure for producer gas.

Briefly, the main points to be considered in an engine designed for producer gas are: First, the size and location of the inlet and exhaust valves and passages; second, the contour and dimensions of the clearance space; third, the provision of an ample and effective ignition gear. If these three major elements of design are in line with the best practice, the question of compression can be very safely left to take care of itself. If, on the other hand, it is impossible or inconvenient with the existing design of parts to secure ample valve area, properly shaped clearances, and a thoroughly good ignition, it is perfectly safe to assume that no amount of compression

* Abstract of a paper on Gas Engine Construction for Producer Gas Use, read before the National Gas and Gasoline (Petrol) Engine Trades Association, June, 1909.

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will make the engine satisfactory for operation on producer gas.

The matter of valves, both as to their location and area, is of great importance. The velocity of gas travel through the ports, both for inlet and exhaust gases, should never be permitted to exceed approximately 80 feet per second. A much lower figure than this will give more satisfactory results. If we assume the customary piston speed of 700 feet per minute we might arbitrarily establish the clear internal diameters of the inlet and exhaust valves at approximately .45 times the cylinder diameter. The cages and passageways leading to the valves proper should also be of extremely ample dimensions, particularly the exhaust passages. Not only should the piping and connections be of ample area, but they should also be provided with very easy curves, and all sharp turns or angles in the exhaust manifold connection should be studiously avoided.

The location of the valves necessarily varies with the type of engine. In general, however, the location of the valves has a very notable effect on the shape of the clearance or combustion chamber; for the highest efficiency the clearance or compression space in the end of the cylinder should have a minimum of exposed heat-absorbing surface in proportion to the volume of the charge. This means that the valve should by all means open directly into the combustion space, and that all pockets or any other notable irregularities of the contour of the combustion chamber should be discountenanced. Engines for use on producer gas must never have any small communicating openings that do not close up flush with the internal surface of the combustion space. Relief cocks and air-starter connections, as well as taps for indicator connections, priming cocks, etc., must all be fitted with plugs or valves closing down flush with the internal surface of the combustion space. Failure to give careful attention to this feature will invariably result in very annoying prematures and back firing.

The question of ignition cannot be given too careful attention. Not only should the mechanism be perfectly reliable, but the source of current must be such as to give a very unusually heavy and hot spark. Ignition devices suitable for use on gasoline (petrol) are frequently entirely unsuited to producer gas. Electric ignition should, of course, be utilized exclusively, and the writer is personally very much inclined to favor a thoroughly substantial form of make-and-break igniter, either mechanically or electrically operated. Electrically-operated igniters of this kind should be supplied from a current source giving not less than 40 E. M. F., and capable of supplying from 2 to 5 amperes of current. For mechanically-operated make-and-break igniters a smaller current supply will suffice. Ordinarily, a storage battery of 10 to 12 volts E. M. F., and a fairly low internal resistance, will give entirely satisfactory results. Dry batteries and primary batteries of any ordinary type are not well adapted to this service, and storage batteries for ignition purposes, together with small dynamos for charging same, are worked out to a degree of perfection that makes the use of any other form of ignition a needless experiment.

The best results are secured when the point of ignition is located very close to the geometric center of the compressed charge. On cylinders over 12 or 14 inches in diameter the question of multiple ignition is worthy of careful consideration.

Coming to the matter of speed regulation, producer gas lends itself to several different methods of governing, but it is by all means best to adopt some form of throttling governing by which an impulse is secured on each regular stroke. Probably the best method is that which involves the throttling of the quantity of the discharge with constant quality of mixture by limiting the lift of the gas inlet valve. Governing by varying the quality of explosive mixture with constant compression can also be worked out in a very satisfactory way for producer gas. The success of this method is due largely to the extremely wide variations of mixture which are explosive when producer gas is used as a fuel. This method has not been used as much as the former method, but the writer sees no reason why it should not be equally successful.

One point in connection with engine design which usually receives very little, if any, consideration on the part of the average builder is the question of the proper control between the proportion of air and gas where the method of governing by constant quality of mixture is employed. It is of the greatest possible importance, particularly in multiple-cylinder engines, that the quality of the mixture in each of the several cylinders be controlled at one point and that by the single movement of one lever or valve. Furthermore, it is very important that this lever, or valve, should have attached to it an indicating device, showing at a glance the amount of air opening for any given position, since under normal conditions it is customary to run suction producer plants with the gas valve wide open, and control the mixture by closing down to a greater or less extent on the air pipe.

Another question comes up in connection with the remodeling of existing designs that is in some particulars deceptive and apt to lead to disaster. We refer to the strength of the various parts of the gas engine. A producer-gas engine of given cylinder bore and given stroke will develop anywhere from 20 to 30 percent less power than a gasoline (petrol) engine of the same dimensions. It is quite natural, therefore, to assume that the stresses on the various parts when operating on producer gas will not be so severe as when operating on gasoline (petrol). The manufacturer, however, must not lose sight of the fact that the higher compression ordinarily employed with producer gas involves at the same time higher initial explosion pressures, and consequently greater strain on the various parts, than where gasoline (petrol) is used as fuel, even though the mean effective pressure and the consequent power is less. It is, therefore, ordinarily not sufficient to take a gasoline (petrol) engine of given cylinder dimensions and correct only the valve areas, clearance, compression and ignition, to make it suitable for producer-gas work. It is usually necessary at the same time to materially increase the strength of all working parts. These remarks, of course, apply particularly to small engines that are not ordinarily subjected to careful analysis as regards the stresses on the various elements. Larger engines are usually checked over with great care, and remarks of this character as to the strength of the various parts of the mechanism would be entirely out of place.

There is not space within the limits of a brief discussion of this kind to take up in detail all of the various features of design that contribute to success in producer-gas work. We can only casually mention the various questions that can be raised in regard to such matters; for example, the cooling of the various parts of the engine. There is room for a great deal of discussion on the question of water-cooled valves and cages and water-cooled pistons. Effective jacketing is at all times a matter of importance, but thorough cooling of every portion of the engine is of double importance where high compressions are employed, as is the case with all producer-gas work.

The question of ignition control is another matter that involves a great deal of possible discussion on both sides. It is, of course, always essential to provide means by which the time of ignition can be advanced and retarded while the engine is in operation, and it is preferable that the exact point of ignition be indicated by the position of the mechanism. There is also a strong tendency on the part of many builders to advance and retard the spark for different loads, to compensate in part for the variation in flame propagation, due to change either in quality or degree of compression of the mixture.

We wish to take occasion at this time, however, to mention, casually, one producer bug-a-boo that has been the cause of

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considerable anxiety in the past, but which we believe to be at the present time strictly within the class of those things that are thoroughly understood. We refer to the question of hydrogen content for producer gas. Several years ago every manifestation in a producer-gas layout that was not thoroughly understood was attributed to the presence of hydrogen in the gas. It has probably been held responsible for more different kinds of trouble than any other single element, and has in all probability been responsible for few, if any, of the various deeds of evil that have been laid at its door. We will take pains at this time to enumerate a few of these features, as it is quite possible that they may come up again from time to time to cause aggravation and annoyance, and perhaps throw discredit on producer-gas power.

Several years ago, back-firing was a very common fault of producer-gas operation, and this was almost universally attributed to excess hydrogen. It has, however, been fully demonstrated that hydrogen in the gas is never responsible for back-firing. This condition may be produced by a number of defects, chief among which we would mention the presence of unplugged indicator and relief cock openings communicating with the combustion space, faulty location and timing of the ignition mechanism, or the overheating of either metal surfaces or carbonized oil within the combustion space. Pre-ignition of the compressed charge is another matter for which hydrogen was long held responsible. It was maintained that owing to the extremely inflammable nature of hydrogen, it would frequently ignite during compression, causing excessive pounding and other similar unpleasant manifestations. The writer has yet to observe a single authentic case of pre-ignition that was due to the presence of hydrogen. We have personally observed the operation of engines on which the compression was run up experimentally to as high as 220 pounds per square inch when supplied with gas containing 27 percent free hydrogen, without any sign of pre-ignition. On the other hand, we have had violent pounding on engines with not over 125 pounds compression, when operating on gas containing not to exceed 12 percent hydrogen. When we consider that the ignition temperature of hydrogen is only a little over 100 degrees lower than that of carbon-monoxide, and that if we assume the most unfavorable condition, namely, a temperature on the charge at the beginning of the compression stroke of 212 degrees F., and strictly adiabatic compression with no loss of heat, a compression of something over 300 pounds per square inch would be required to reach the ignition temperature of hydrogen. The improbability of pre-ignition from this source is immediately apparent. One characteristic of hydrogen, however, must not be lost sight of, namely, the fact that flame propagation is very much more rapid through hydrogen than through carbon-monoxide. In other words, an explosive mixture containing a high percentage of hydrogen will be completely burned in a very much shorter time after the occurrence of the igniting spark than would be the case where the hydrogen percentage is low and the combustible gas largely carbon-monoxide. A sudden variation in the hydrogen in producer gas may be the cause of violent pounding, not through any spontaneous ignition of the charge from compression, but on account of the more rapid rate of flame propagation having the effect of abnormally advancing the time of ignition, the resulting pounding being due to the fact that the igniter is set quite too early for the correct ignition of a mixture with so high a rate of flame propagation. It may be stated, therefore, as a fairly well-established fact, that a properly designed gas engine will handle producer gas with any possible continuous percentage of hydrogen without any irregularity in operation whatever, and that whatever irregularities may be traced to the hydrogen content of the gas are due rather to the variation in percentage of hydrogen than to the actual amount present.

TRIALS OF THE JAPANESE ARMORED CRUISER IBUKI.

The Japanese armored cruiser *Ibuki* has recently completed a series of very successful steaming trials at the Kure Navy Yard in Japan. This vessel is equipped with Curtis marine reversible turbines, built by the Fore River Shipbuilding Company, Quincy, Mass., which were shipped to Japan and installed in the vessel at the Kure yard.

The turbines drive twin screws, and are of 12 feet pitch diameter, with seven stages. They were guaranteed to deliver 21,600 brake-horsepower. The table given below shows the results obtained on the various trials.

The contract guarantees were made on a basis of 250 pounds steam pressure and 28 inches vacuum at 200 revolutions for two-fifths power and 255 revolutions at full power. The actual trial conditions were somewhat under these, and the corrected water rates in the table are to allow for the differences.

The reversing power of the turbines was tested at the end of the four-fifths power run by running astern for fifteen minutes, keeping the same firing interval and conditions in the boiler room as were used in going ahead. The turbines ran reversed at 186.3 revolutions, and developed 11,035 brake-horsepower. Also the general maneuvering qualities of the vessel were excellent.

The Fore River Shipbuilding Company have also supplied Curtis turbines for the battleship *Aki*, and Japanese yards are building Curtis turbines for two other battleships and three scout cruisers; two of the latter being ordered as a result of the successful outcome of the *Ibuki* trials.

Each of the turbines of the *Ibuki* has seven ahead wheels and two reverse wheels, all in one casing, and each in a compartment formed by diaphragms inside the casing. The first ahead-wheel and each of the reverse wheels has four rows of moving buckets, while the remaining wheels have three rows each. The steam leaving any wheel is directed through nozzles in the diaphragms onto the buckets of the next wheel. The turbines are reversed by simply shutting off steam from

TABULATED RESULTS OF OFFICIAL TRIALS OF THE JAPANESE CRUISER IBUKI.

	1/5 Power.	2/5 Power.	3/5 Power.	4/5 Power.	Full Power.
Duration of trial, hours	8	8	24	6	6
Steam chest pressure gage	221	228	230	240	239
Quality of steam	Sat.	Sat.	35° Sup.	28° Sup.	53° Sup.
Exhaust shell vacuum, inches	28.1	27.5	27.2	26.4	25.7
Revolutions per minute	151.2	189.1	215.7	235.5	250.5
Brake-horsepower	5,077	10,077	15,730	20,978	27,142
Water per hour for main turbines, pounds	108,021	183,083	256,910	330,339	407,987
Water rate per brake-horsepower, pounds	21.27	18.17	16.35	15.73	15.03
Available British thermal units in steam	348.	337.	341.5	329.	325.8
Efficiency of turbines, percent	34.3	41.6	45.5	49.2	52.
Water rate corrected to contract conditions	.....	16.76	.....	.....	13.88
Guaranteed water rate	.....	17.	.....	.....	15.

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the ahead steam chest and opening the valve to the astern steam chest.

The above turbines of the *Ibuki* are similar to those on the United States scout cruiser *Salem*, except that their pitch diameter is 2 feet greater.

The Operation and Management of the Parsons Marine Steam Turbine as Practiced on the U. S S. Chester.*

BY LIEUTENANT A. F. H. YATES, U. S. N.

The turbine installation on the *Chester* is of the standard type designed by the Parsons Marine Steam Turbine Company. There are six ahead turbines, a high-pressure cruising, intermediate-pressure cruising, two main high-pressures, and two low-pressures. There are four independent shafts, each fitted with a single, solid, three-bladed propeller of 6 feet diameter and 6 feet pitch. Reversing turbines are incorporated into the exhaust ends of each of the low-pressure turbines. A reference to Fig. 1 will make clear the arrangement of turbines on shafts and the piping leads. The high vacuum maintained is assisted by the use of the Parsons vacuum augmenter. Independent air pumps are installed, with connections to the condensers through water seals. (Fig. 2.) The vacuum augmenters are virtually steam jets so arranged as to siphon the air from the condensers at a point just above the water level and deliver it to the water seal for the air pumps to handle, at a pressure of from 1 to 1 3-10 inches of mercury higher than the condenser pressure. The discharge from the siphon passes through a small condenser for condensation and reduction in temperature. The system of lubrication installed provides for the supply of oil to thrust bearings, main bearings and spring bearings at a pressure of about 10 pounds. The oil is supplied by steam-driven oil pumps and passes first through coolers on its way into the supply main. Separate pipes from the main pipe to the different bearings conduct the oil to the latter, and separate pipes take it away by gravity to common return mains, which discharge into the oil tanks in the bottom of the engine rooms. It is from these tanks that the oil pumps take suction. There are no valves in the system. The pipes which conduct the oil from the bearings have sight glasses in them, so that the flow of oil may at all times be under observation, and at the same points thermometers are attached to indicate the temperature of the oil as it leaves the bearings.

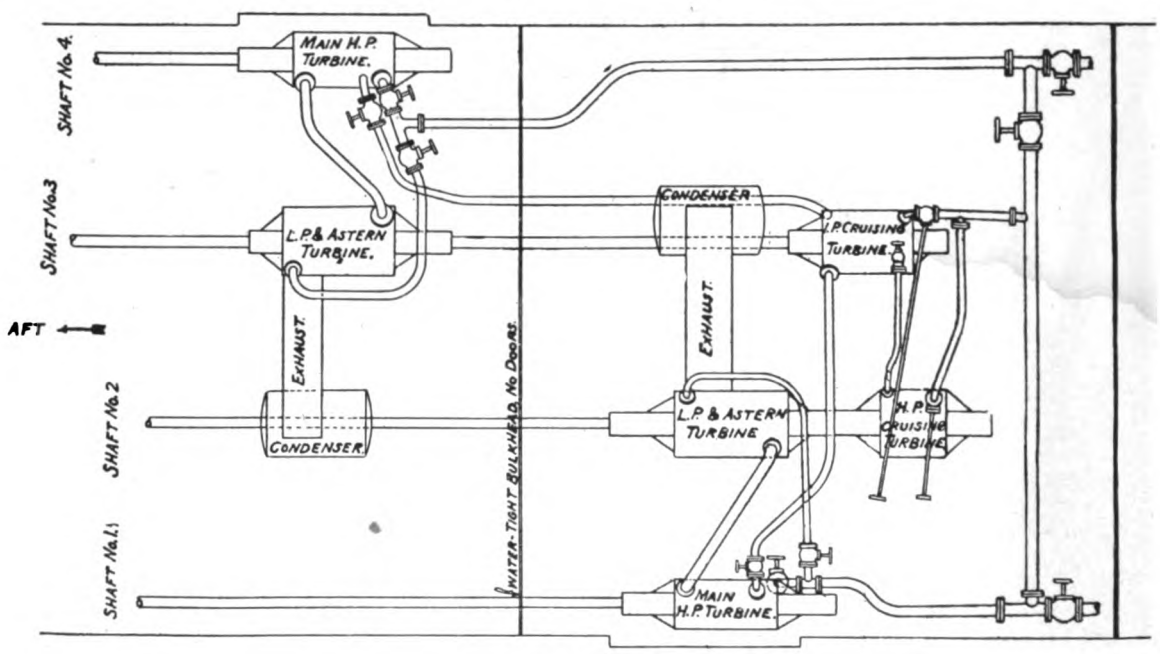

FIG. 1.

WARMING THE TURBINES FOR GETTING UNDERWAY.

The length of time necessary to properly prepare the turbine for trial is greater than for the reciprocating engine. Three hours and a half are usually allowed on the *Chester* from the time of beginning to warm up to the time of getting underway, though the turbines are generally tried and reported ready after 3 hours. They have been made ready in 24 minutes, and it is believed that they could be prepared in 10 minutes. That which is to be avoided is local heating and consequent uneven expansion. Steam is admitted to a circular steam belt, and in consequence the steam takes its initial direction over all parts of the rotor drum at the same time, and toward the exhaust end. At the exhaust end the steam leaves the turbine at the one point on the side of the casing where the exhaust pipe is connected. For this reason it might be expected that the near side would benefit by steam of perhaps somewhat higher temperature than that on the far side. In order to counteract any ill effects which might result from such cause, it is considered advisable to utilize the jacking gear while warming up, turning the shafts one-quarter or one-half a cycle every half hour. In order, too, that the greatest possible benefit may be derived from the heat of the steam, it is considered a wise practice to keep an extremely low vacuum, moving the air pumps as slowly as possible with safety. The various exposed parts of the turbine not covered by lagging should be felt with the hand from time to time, as much can be learned in this way as to the uniformity of heating. The use of auxiliary exhaust steam for warming up was at one time considered, but the idea was abandoned.

Steam should be placed on the glands at the same time it is admitted to the turbines.

The general procedure of warming up the turbines on the *Chester* is as follows:

1. Open the drains of all turbines to the condensers and unseat all throttle valves and spring-loaded non-return valves.

* Abstracted from the *Journal of the American Society of Naval Engineers.*

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2. Open main injection valves and overboard delivery valves. Start main air and circulating pumps, having circulating connections open to the vacuum augmenter condensers.

3. Crack the cross-connection valve in the communicating pipe between the auxiliary steam line and the main steam line in engine room, thereby putting steam on so much of the main steam line as is in the engine rooms.

4. Close turbine throttle valves and open throttle by-pass valves. Keep a low vacuum.

5. Regulate pressure on auxiliary exhaust line, opening its connection to the steam-gland system sufficiently to give a pressure on the glands of about 1 pound.

6. Examine the strainers on oil pumps and test both pumps. Note first the amount and condition of oil in the gravity-return tanks; draw off any water that may be in the tank. See that water service is on the oil cooler and on the water-jackets of the spring bearings.

7. Steam having been raised on the additional boilers to be put in use, connect them, and open the engine-room bulkhead stop valves, closing the cross-connecting valve between the main and the auxiliary steam lines. Close throttle by-pass valves, if desired, and continue warming up through the throttle valves partially unseated.

8. The turbines being warm, a set of micrometer readings for dummy clearances should be taken and entered in the log.

9. Open valves admitting steam to the vacuum augmenters. Disconnect jacking gear.

10. Start oil pumps, by-passing the oil around the cooler. It should continue to be by-passed until the ship is underway, and the oil has reached a temperature of 100 degrees F., otherwise it will be sluggish and fill the bearings with difficulty.

11. Speed up the air and circulating pumps and raise the vacuum, and when a good vacuum has been obtained, report the turbines ready for trial. Keep all drains open until well underway. When lying at anchor with banked fires, and with orders to keep the turbines warm, they should be jacked over at least once an hour. They remain warm a long time after securing, but steam should be kept on them. Fifteen minutes' notice is then considered sufficient.

TRYING THE TURBINES.

The turbines are tried separately at a low speed under light pressure, and if properly warmed they will move easily, and the chances of their giving trouble when opened out are none. In trying them "ahead slow," sufficient pressure should really be given the main high-pressure turbine to insure the turning of its accompanying low-pressure turbine by the exhaust of the former. This will not happen if the exhaust pressure is extremely low. The test of the backing turbines is frequently repeated several times, it being the case very often that a rather high-pressure is needed to turn them the first time. The cruising turbines should, of course, be tried out. These turbines are so much smaller than the others that they are almost certain to be warm if the others are, and no difficulty has ever been experienced with them. If there is any interference of the blading in any turbine, it will make itself evident by an intermittent rumbling or grinding noise, which, if very serious, will be very shrill. The ears should be strained when the turbines are first tried, in order to detect even the slightest noise, as a slight interference at such slow speed might increase and give trouble upon opening out the turbine. If at any time there is a reasonable doubt as to conditions, the turbine should be stopped promptly and given more time to warm up. No more than 15 or 20 minutes are apt to be lost by so doing.

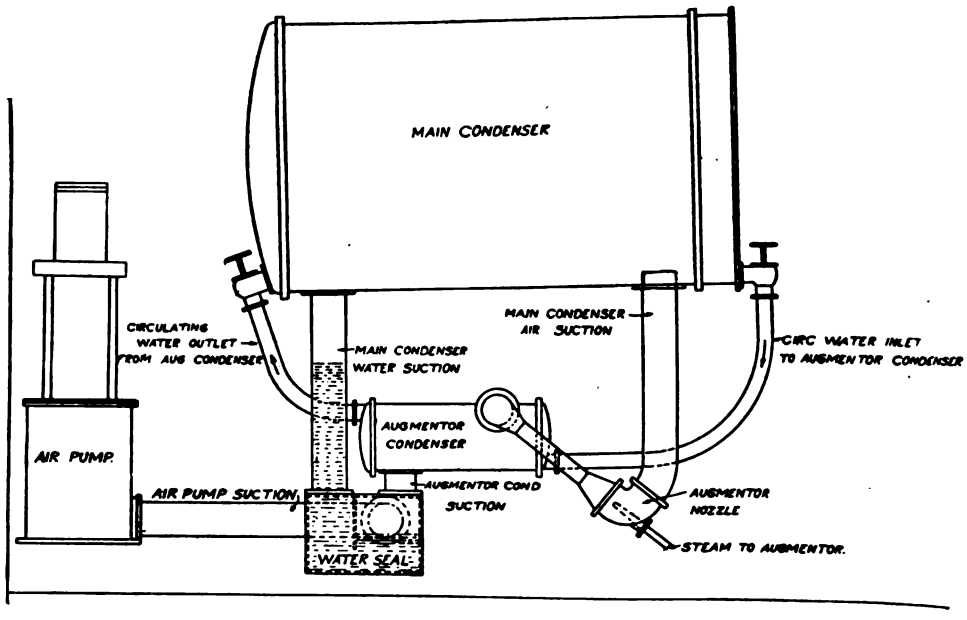

FIG. 2.

MANEUVERING, STOPPING AND STARTING.

Maneuvering is accomplished with much greater ease than with the reciprocating engine, as there is no reversing gear necessary to operate or to fail of operation. The operation consists simply of the opening and closing of valves, and consequently but very few seconds are ever necessary. The mistaken idea held by some who are not familiar with the turbine, to the effect that maneuvering is a lengthy operation, is entirely without foundation.

Full steam pressure may be admitted to the turbine as fast as the throttle can be opened, and the time elapsed to full open is but a few seconds. The valves should, most naturally, not be spun open with lightning speed, neither should they be opened irregularly by jerks, but they may, with perfect safety, be opened smartly and with precision, and a sluggish response to a signal is entirely unnecessary. During periods of rest it is a good idea to admit an occasional blast of steam into the astern turbines for the purpose of keeping them warm. They act more efficiently when warmed up thoroughly, and the admission of light blasts of steam can be accomplished without turning the shafts. Due to the fact that the astern turbines

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are on the same spindle as the low-pressure ahead turbines, the dummy strips on the former must be of the radial type, whereas the latter are of the contract type. Fig. 3 illustrates the two types. The latter type permits of close and comparatively uniform regulation of the clearance. The former type is a permanent adjustment and cannot be as efficient, as it must permit of more elastic limits. But one of the two dummies can have the type suited to adjustment, and it is, of course, most important that the ahead turbine should

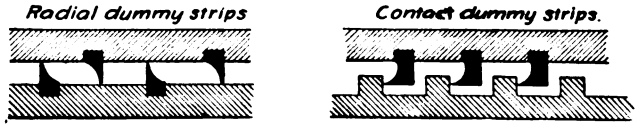

FIG. 3.

be the one so fitted. It is because of the type of dummy strips fitted in the astern turbines that more careful handling of these turbines is considered necessary; injudicious handling of the astern turbine throttle valves is particularly to be avoided.

If a reverse signal is received from full ahead to full astern, no anxiety need ever be felt. The valve to the astern turbine may even be started open while the ahead throttle is being closed, and, as the combined act requires so little time, the reversal may be considered as almost instantaneous.

If running under cruising turbines and a reverse signal is received, the execution of the signal involves no greater delay. Indeed, even less care is necessary than if the ship were running ahead under the main high-pressure turbines. This is because spring-loaded, non-return valves are fitted in the exhaust pipes, between the intermediate-pressure cruising turbine and each of the main high-pressure turbines. Steam is promptly cut off the cruising turbines, and, at the same time, the astern turbines' throttles are opened. Delay in attending to the former would cause no embarrassment, however, because of the valves above mentioned, which close automatically. In actual practice the non-return valve is usually closed by hand at the same time the remainder of the operation is being attended to. This should clear up any doubt as to occasion for accident, due to the number of turbines in operation.

Water in the turbines, due to condensation, or as a result of priming, need cause no apprehension. The drains are always open when maneuvering, and if the presence of water becomes known, it disappears almost as quickly as it comes. It is broken into small particles and passes out with the exhaust steam in the form of spray, causing no damage.

OPERATION WHEN FAIRLY UNDERWAY.

Reference is again invited to Fig. 1, in connection with a brief description of the manner in which the turbines are used for economical performance at different speeds.

First, for low speeds—up to about 18 knots—the steam passes through all six ahead turbines, both the high-pressure cruising and the intermediate-pressure cruising being connected up with the four main turbines. Steam admitted to the high-pressure cruising exhausts into the intermediate-pressure cruising, and from the latter it exhausts through separate exhaust pipes to each of the main high-pressure turbines. From the latter it exhausts into the low-pressure turbines, thence into the main condensers.

Second, for moderate speeds—up to about 23 knots—the steam passes through five ahead turbines; being admitted to the intermediate-pressure cruising turbine it exhausts, as before, into the main high-pressure turbines, thence into the low-pressures and into the condensers. The high-pressure cruising turbine revolves idly in a vacuum, its drains being open to the condenser.

Third, for highest speeds, only the four main turbines are used, steam being admitted to each main high-pressure turbine independently, and exhausting in each case through the connected low-pressure turbine and into the condenser. Both cruising turbines revolve idly in a vacuum, their drains being open to the condenser.

Reduction of power in each of the arrangements is obtained by throttling. Increased power is obtained by raising the pressure in the steam belt of the turbine which is being used as the initial stage. In the first arrangement, additional power may be secured through the use of a by-pass valve from the first to the second expansion, and also by admitting live steam direct to the intermediate-pressure cruising or to the main high-pressures. Similar increase in power may be obtained in the other arrangements. Ordinarily these alternatives are not used, though they are apt to be of benefit at any time.

The turbines should run without noise and without vibration. Such noises as have been mentioned above, if heard while the ship is underway, would indicate unusual conditions, which should be investigated immediately. As long, heavy turbines may whip slightly in a sea-way, due to the rolling of the ship or the racing of the screws, this might cause the blades to rub now and then and the turbine to groan or vibrate slightly. If the clearances of the dummies were very scant, the dummy rings might run a slight risk of grinding for similar reasons, but the likelihood of such an occurrence is very remote. Any grinding at this point would indicate too scant a clearance, and would be very serious, necessitating readjustment at first possible opportunity.

It would appear that greater readiness for service should be expected of the turbine than of the reciprocating engine, as overhaul is so seldom necessary and usually of a mild nature even then. Hardly any attention need be paid the turbine proper once it is underway, and no injury to attendants need ever be expected. The advantage resulting from the purity of the steam and its freedom from oil extends to the boilers, condensers and feed-water heaters. Barely any cylinder oil is used on the auxiliaries in the engine room of the *Chester*, a mixture of graphite and kerosene proving satisfactory for these engines.

In smooth weather a constant initial pressure, with a steady vacuum, corresponds to a constant speed, but varying conditions of wind and sea, or great variations in boiler pressure, have a detrimental effect in this respect. So long as the boiler steam is dry no variation in initial pressure for a constant speed is necessary, but when, with a drop in pressure, the steam becomes wet, the effect is felt at once. For this reason the fire-room force must be trained especially to preserve fairly constant limits of pressure. The effect of wind and sea is really marked, and higher pressures become necessary for the same speed, with constant loss in economy. This is against the turbine, but in spite of it the turbine ship is admirably fitted for heavy-weather work. The turbines do not labor or require throttling, but run with ease, and adapt themselves freely to the changes. It may express their action best to say that they appear elastic under such conditions.

With a sudden change in speed it is found that the turbine does not firmly establish itself to the new conditions for some considerable time. The proper pressure to carry for a specified number of revolutions per minute cannot be exactly determined for nearly an hour, though, of course, one suitable for the approximate speed can be found in a very few minutes; that is to say, within a quarter of a knot and less.

Running on a standard number of revolutions per minute is attended with the difficulty of ascertaining from time to time how many revolutions are being made.

Once the speed is established, however, it is maintained constant with comparative ease, though the man at the throttle must ever be at his station with his eye on the pressure gage,

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The *Chester* picks up headway with lightning rapidity. While this, to a large extent, is due to the fineness of her lines, it is also due to the turbine installation. Maneuvering from rest should be harder with a ship having small propellers, but the turbines may be given steam so quickly and they reach their speed so quickly, that I believe for quick ahead action they are, on the whole, superior. I would not like to say the same for astern motion, however.

The two most essential points in connection with the economical operation of the turbine are undoubtedly dry steam and a high vacuum. The *Chester's* boilers furnish a high quality of dry steam, and no ill effects from this source are experienced as a rule.

As regards a high vacuum, the reason for its importance lies in the fact that the steam undergoes a greater range of expansion in the turbines than in the reciprocating engine. I might say, roughly, that for 1-inch drop of vacuum on the *Chester* her speed will drop off half a knot.

The provisions made for the utilization of auxiliary exhaust steam in the turbines have already been referred to above. There are many different conditions under which the auxiliary exhaust steam might be used, sometimes in one place and sometimes in another, depending upon the relative pressure conditions. The advantage to be derived should, however, be determined before it is used. Under some circumstances, as will be shown later, the auxiliary exhaust is drawn off to feed the gland system. Then again it may be used in the feed heaters. If it is used for these purposes there are not many occasions where it will be found sufficiently plentiful for additional use in any of the turbines. Therein lies a question as to whether greater benefit is derived from its use in the feed heater or in the turbine. I am hardly prepared to speak authoritatively in reply thereto. Benefit is derived in each case. Conditions have been noticed on the *Chester* where its application to the turbines resulted in a half knot increase in speed; whereas, on the other hand, its use in the feed heaters would have provided feed water of a temperature of 230 degrees F. or 240 degrees F.

The gland system on the turbines is one which requires more or less constant attention. As installed on the *Chester* it consists of a main pipe, which passes the entire length of the two engine rooms, forward and after. Branch pipes connect it with each gland of all the turbines, twelve in all. The only valves in the system are on the main pipe. There are two cutout valves, one to isolate the portions of the main pipe which lie in the respective engine rooms, to permit of independent management, and one to isolate the extreme end of the main pipe which lies abreast the cruising turbines. Neither valves are ever closed, as the gland system is worked as one unit and independent management in the two engine rooms not attempted; the one for isolating the cruising-turbine gland system would only be used in the event of those turbines being disconnected from the main shafts. There is, therefore, free communication in the system between all glands. The only other valves are one in each engine room for the admission of auxiliary exhaust steam to the main pipe, and a valve in a branch pipe to the condenser with which to relieve any excessive pressure on the system. In order to thoroughly understand the management of the system one must first have a clear understanding of the functions of the glands, especially as these functions are dependent upon existing conditions. The steam gland is, of course, the counterpart of the stuffing box on the reciprocating engine, and is intended to prevent leaks. While some of the glands tend to blow steam, others tend to suck air. The amount of steam leakage from glands under internal pressure depends upon the efficiency of the same and cannot be entirely avoided. The effect of the leakage of air through the others into the turbine will be noticed by a drop in the vacuum. As was stated above, the glands are all in free internal communication through the gland-system piping, so the leak-off steam from glands under internal pressure finds its way to the other glands and has the effect of sealing them against air leakage. Some of the steam leakage enters the atmosphere, but that cannot be avoided with the type of gland fitted. The amount of leak-off steam which enters the system is not always sufficient to seal the other glands, and in such cases auxiliary exhaust steam must be used in addition. On the other hand, the leak-off steam is under some conditions excessive, and it is then that the gland relief valve is opened to draw off the surplus to the condenser. About 1 pound steam pressure per gage is ordinarily used on the gland system, with good results.

In view of the high speed at which the shafts revolve, a constant and vigilant watch must at all times be kept on the bearings and on the oil system. Care of the bearings and of the oil service equals in importance the skillful warming up of the turbines for service. The main bearing journals are so constructed that, in the event of the babbit melting, certain raised portions of the brass will take the weight of the shaft during the time necessary to slow down and stop. This is in order to save the stripping of the blades, which otherwise would be a natural consequence. The use of the hose on the bearing casing when a hot bearing develops is a useless alternative, though after the turbine is stopped it can do no harm to reduce the temperature. Sight glasses are fitted in all of the branch discharge pipes from bearings, and a glance at these shows whether or not the flow of oil is normal. Thermometers are also fitted at these points, and the temperature of the oil discharged from each bearing is readily observed. The oilers in making their rounds should not be guided entirely by these devices, but should feel each bearing with the hand. No calculations have been made on the amount of oil required for operation, but the tremendous oil-saving with the use of turbines is most apparent. A precaution that should be observed is the daily inspection of the gravity-return oil tanks when in port for the presence of grit, dirt or an excessive amount of water. The water can be drawn off, but if it is excessive, steps should at once be taken to ascertain where it comes from. It should not come from the coolers, on account of the oil pressure being greater than the water pressure, but this is one place that should at least be inspected. Another place is in the water-jackets on the spring bearings. The water should, of course, be tasted first, to determine whether salt or fresh. It is surprising how much water may be expected from sweating.

The marked effects of cavitation supposed to exist when high power is reached exist more in fancy than in fact. The second standardization trial of the *Chester* conclusively proved that she had made all speeds that had ever been claimed for her. The efficiency of the propellers falls off at the higher speeds, but not to an alarming extent, and no evidence of their breaking down or of erratic behavior has ever been detected. New propellers of even greater disc area are to be fitted later, and slightly greater speed is expected.

My remarks on operation must be brought to a close after mention of the following points to be observed when securing the turbines after coming to anchor. Steam should be tightly shut off the glands and off the vacuum augmenters. Special precautions should be taken to see that water service is kept on the vacuum-augmenter condensers, however, and the main air and circulating pumps should be run for about 3 hours after steam is shut off the turbines. The latter is necessary, as hot vapor remains in the turbines and develops for a considerable time after the steam and water has been drained out, and the condensers might be damaged if secured too early; even after securing, condensation continues for a considerable length of time, for the turbines remain more or less warm from 12 to 24 hours, depending on the surrounding temperature.

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THE MARINE STEAM ENGINE INDICATOR—III.*

BY LIEUT. CHARLES S. ROOT, U. S. R. C. S.

THE SCOTT-RUSSELL MOTION AND MECHANISM DERIVED FROM IT.

If the reader will provide himself with a flat carpenter's square, a thin batten, a pencil and a couple of round hardwood points, he will have the means for graphically analyzing this motion in a most interesting manner. Drill three holes on the center line of the batten, *A* and *E*, near the ends and *C*, midway between them, so that *AC* equals *CE*, as shown in Fig. 21. Insert the sharpened hardwood points in *A* and *E* and

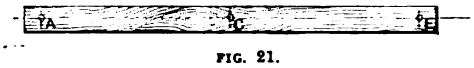

FIG. 21.

a pencil in *C*. See that the points are all accurately in line and equidistant, with the pencil point a little longer than the end points. All three points should be sharpened with the same degree of taper.

Place the square on a smooth sheet of paper, resting on a plane surface. Take the batten in hand, holding the point *E* in the inside corner, and the point *A* against one of the legs of the square, so that the pencil is perpendicular to the plane of the paper, all as shown in Fig. 22. Now, keeping the

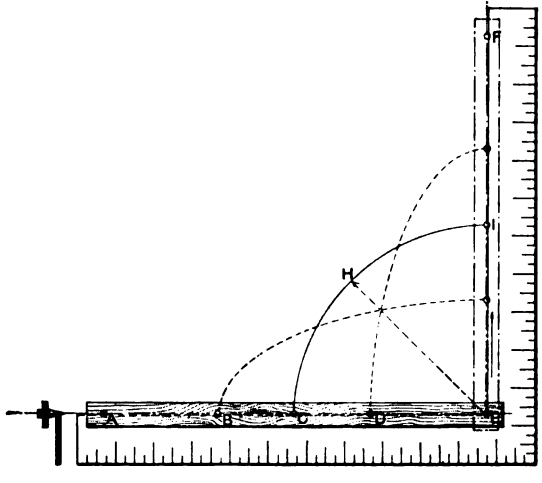

FIG. 22.

wooden points *A* and *E* in contact with the square, move the point *E* in the direction of the arrow until it reaches *F*. The batten will then occupy the position shown in dot and dash lines. During the movement described the pencil will have traced the arc *CHI*. This arc will be truly circular, with a radius *HE*, the center being at *E*. If now the pencil be removed to any other hole on the center line of the batten, as *B* or *D*, and the movement repeated, as before, the arcs traced will be elliptic in every case, as shown by broken lines in the figure.

Consider the slider-crank mechanism shown in Fig. 23. This is the Scott-Russell parallel motion proper. The pin *C* is in the middle of *AE*, and the link or crank *CG* is equal in length to one-half of *AE*. Here, the point *A* is constrained to move in the straight line *AG*, as in Fig. 22, and the point *C* in a true circular arc by the crank *CG*. Likewise, every other point on *AE*, except those at the ends, will describe elliptic arcs, as shown at *BB'* and *DD'*. The marking point is at *E* and draws the accurate straight line *EG*, which corresponds with the leg *FE* of the square in Fig. 22.[3] The similarity of the two motions will be readily understood without further description.

In order to adapt this mechanism to the indicator and avoid introducing mathematical errors, the arrangement shown in Fig. 24 might be used, the pencil lever *AE* being offset between *C* and *E* to allow the scribing point to pass the fulcrum *G*. With this arrangement the pencil or scriber path *EF* will be an accurate straight line normal to the line *AG*, as has heretofore been demonstrated. The piston rod, whose center line *PP'* is parallel to the pencil path, is connected to *AE* by the slotted

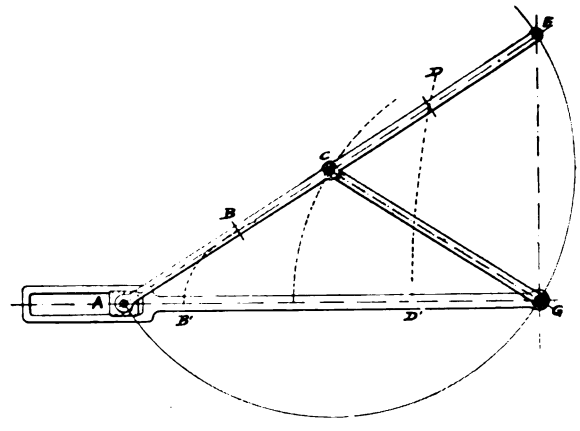

FIG. 23.

crosshead and pin at *B*, the rod being guided in its rectilinear path by the sliding pair at *D*, and by the cylinder and piston. As the distances x and y, measured parallel to *AG*, are always proportional,[4] the velocity ratio $\left(\frac{x+y}{x}\right)$ between the piston and pencil will be constant in all positions of the mechanism.

This movement, although mathematically exact, is not considered mechanically practical, and, so far as we know, has

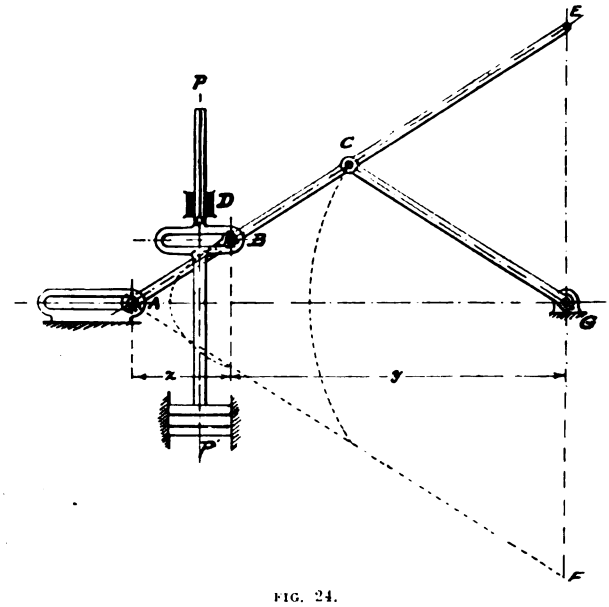

FIG. 24.

never been actually used on any indicator. A link turning on pin joints is always used in place of the sliding pair at *A*, and the piston rod is connected to the pencil lever in the same manner. While this makes a better mechanical arrangement,

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[3] The points *AGE* lie in the semi-circular arc whose center is at the movable point *C*, and this holds true in every position of the mechanism. As every angle inscribed in a semicircle is a right angle, *EG* is always a straight line passing through *C* and normal to *AG*.

[4] By the theorem of proportional triangles.

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mathematical errors are at once introduced. A pencil mechanism, fitted with the substitutions mentioned, and which most closely resembles the arrangement in Fig. 24, is shown in Fig. 26.

For convenience of reference we shall hereafter call the link *AB* (Fig. 26) the "back link," *CD* the "piston rod connecting link," *EF* the "front link," *AP* the "pencil lever," *xx'* the "horizontal axis," and 1-5 the "pencil axis."

In the mechanism of Fig. 26, the front link is half as long as the pencil lever, and is paired to the latter at its middle point by a pin joint, as in the Scott-Russell motion; but the pencil is

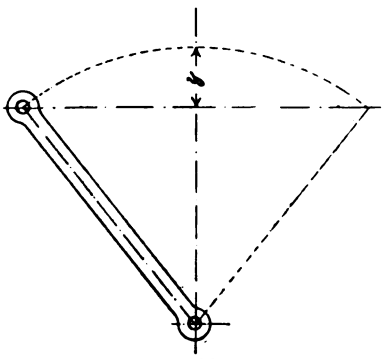

FIG. 25.

caused to deviate from the accurate straight line of that motion by the versed sine *y* (Fig. 25) of the back link. This link (*AB*, Fig. 26) is so located that the versed sine is inclined and the scribing point *P* crosses the pencil axis at 1, 2, 3, 4 and 5, the intervals being equal. The actual pencil path is much exaggerated in the figure in order to show its general form. The piston rod link *CD* is parallel to the back link when the mechanism is in the position shown, and the points *B*, *D* and *P* are in the same straight line. It follows, from what has been said of the pantograph, that whenever the

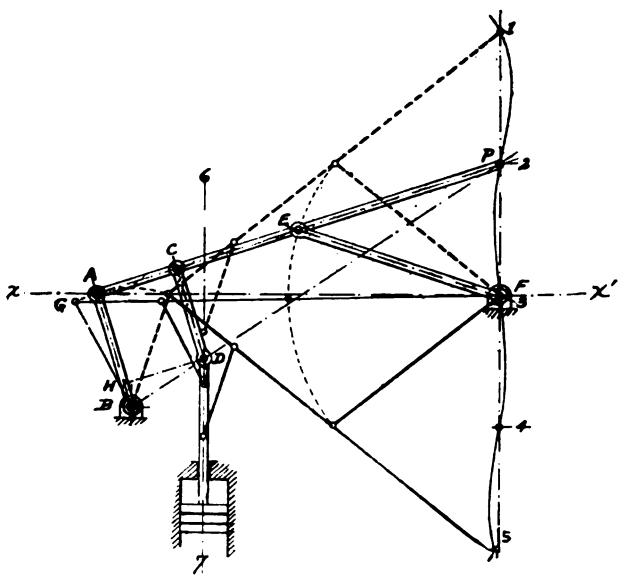

FIG. 26.

points *P* and *D* are on the parallel lines 1-5 and 6-7, the front and back links will be parallel, *AC* and the imaginary link *HD* will be parallel and of equal length, and the velocity ratio between the piston and pencil will be constant. Whether the front and back links are in proper adjustment proportionately on the five points is seen to depend on the accuracy with which the piston follows the line 6-7, which is parallel with 1-5. The accuracy with which the pencil follows its designed path depends entirely on the front and back lines, and is independent of the piston-rod guide. As the mechanism is absolutely correct at five equidistant points on the pencil axis, it is believed to be one of the most accurate of all the approximate motions. The heavy, broken lines show the mechanism in its highest position with the pencil at 1. The fine, unbroken single lines indicate the position of the links

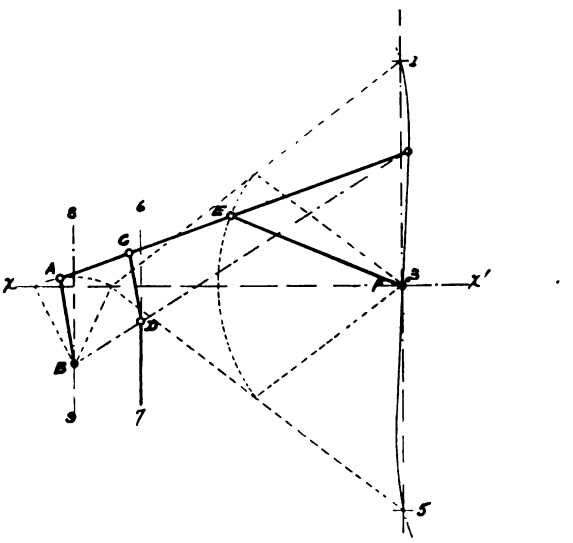

FIG. 27.

with the pencil at 3, and the fine double lines the lowest designed position of the mechanism with the pencil on 5.

Referring again to the back link: If this link be located so that the versed sine falls on the line 8-9 (Fig. 27) and is entirely above the horizontal *xx'*, the pencil will cross the pencil axis three instead of five times, and will have but three points where the movement is absolutely correct. Locations of the back link, intermediate to those shown in Figs. 26 and 27, will result in "five-point" mechanism, but the crossing points will not be evenly spaced.

This mechanism is still further altered in many designs by

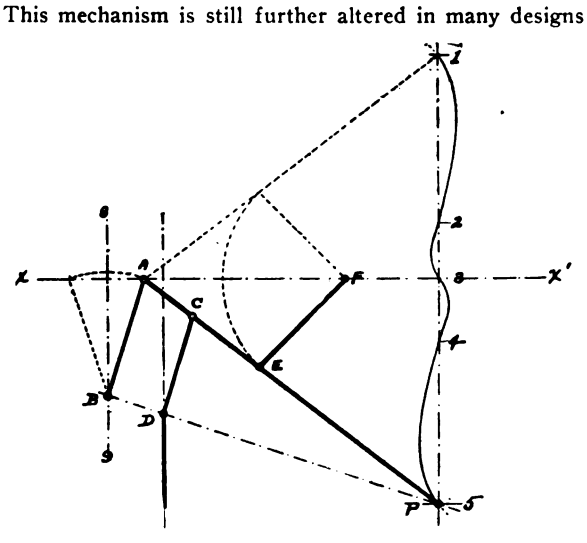

FIG. 28.

shortening the front link and locating it at a greater distance from the scriber end of the pencil arm, as shown in Fig. 28. The path of the point *E* should be elliptic, but it is constrained to move in the arc of a circle whose center is at *F*. As the difference between the circle and ellipse at this point and within the limits used is small, the error introduced is inconsiderable. The characteristic pencil line traced by this mechanism is shown in exaggerated form in the figure. In some instruments, even the location of the point *D* on the line

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BP has been changed, and *CD* made as long as possible by locating *D* close to the piston, the arrangement being similar to that formed in "trunk" engines.

In another form, the front link is paired with the piston rod connecting link at *E*, as shown in Fig. 29. If *P* is made to follow the straight line 1-5, the path of *E* is neither a circle nor an ellipse, but a curve of a higher order; it is so close to a circular arc, however, that (within proper limits and on the

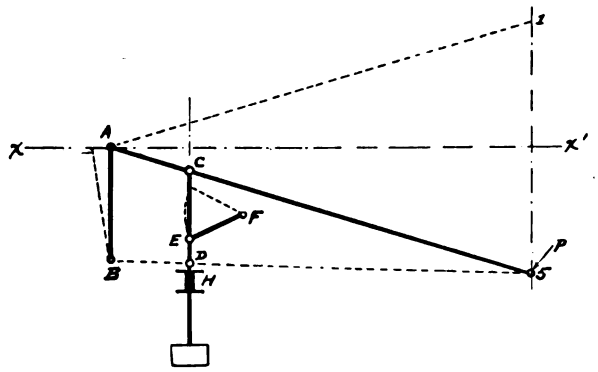

FIG. 29.

scale used) the difference is not measurable. In order that the pencil may follow its designed path in this arrangement it is most important that the piston-rod guide *H* be kept in good order, as any lost motion at this point will cause a large pencil error. Mechanisms in which the front link is made shorter than one-half the pencil lever have a certain advantage, other things being equal, in that the weights have a shorter motion at lower speed than in those instruments fitted with the non-shortened front link.

MECHANISM WITH PENCILS CONSTRAINED BY SLIDING PAIRS.

Several makes of instruments have the pencil mechanism shown in the conventional sketch, Fig. 30. The pencil point

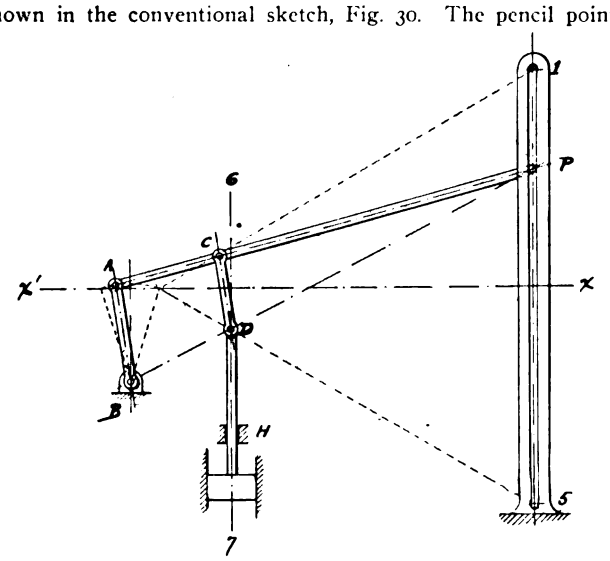

FIG. 30.

P, is guided directly by a straight slide, the links are so proportioned that the pins at *B* and *D* are in line with *P*, and *AB* and *CD* are parallel. These conditions are constant, so long as 1-5 and 6-7 are parallel, and the motion is therefore absolutely correct when properly adjusted. Another well known movement of this sort is shown in Fig. 31. The proportions of the various links are as in Fig. 30. The shape of the curved slot *EE'* is derived as follows: If *P* be guided accurately on the straight line 1-5, the pin *E* will mark out a peculiar curve, which will be neither a circle nor an ellipse owing to the disturbing effect of the back link. Conversely, if a guide be made of the exact shape marked out by *E*, as just described, and this guide be made to constrain the motion of the pencil lever, the pencil *P* will draw an absolutely straight line. From the description of the shape of the guide and an inspection of the diagram, it is seen that this mechanism is absolutely correct, both in its straight-line motion and constant velocity ratio.

Lack of space forbids the description of a greater number of these mechanisms, and, moreover, those described cover the majority of instruments now in use. Many devices ad-

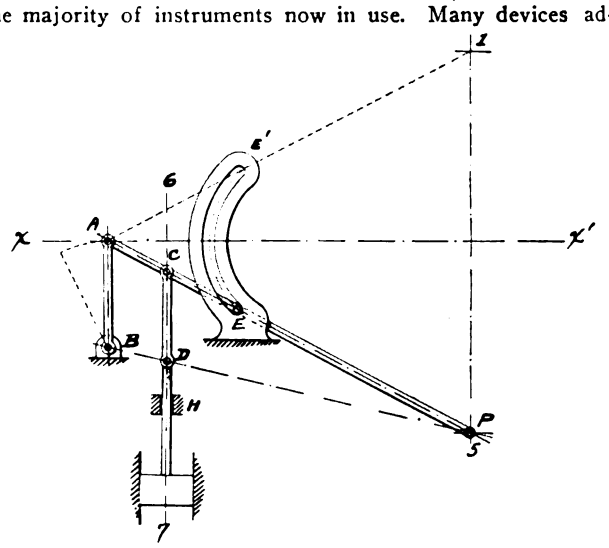

FIG. 31.

ditional to those shown have been made, and there is no difficulty in designing still more of them, but it would be hard to improve on the best of those now in use. The sketches heretofore shown are not to be taken in any sense as detailed drawings, but as conventional sketches, only intended to show general principles.

(*To be continued.*)

Economy of Turbines vs. Combined Reciprocating and Turbine Propulsion.

Sir Christopher Furness, M. P., recently had the machinery of his steam yacht *Emerald* converted so that the three-turbine, three-propeller system was changed to a combined high-pressure reciprocating engine and low-pressure turbine. The following table, published by the *American Machinist*, has been given out by Sir Christopher Furness to give figures for comparison of these two forms of propulsion:

Speed.	Consumption.	Steam Pressure.
	BEFORE.	
10 knots.	18 tons.	150 pounds.
11 "	23 "	145 "
12 "	25 "	140 "
13 "	28 " (?)	...
	AFTER.	
10 knots.	13 tons.	150 pounds.
11 "	16½ "	150 "
12 "	20 "	150 "
13.2 "	22.9 "	150 "

It will be seen that in the first table the steam pressure varied, whereas in the second it was constant at 150 pounds. This makes comparison somewhat difficult, but it is safe to point to the top line in each set of figures; namely, that relating to a speed of 10 knots, when, with a steam pressure of 150 pounds, the consumption figures are 18 and 13 tons. Thus, with the converted machinery the steam consumption was

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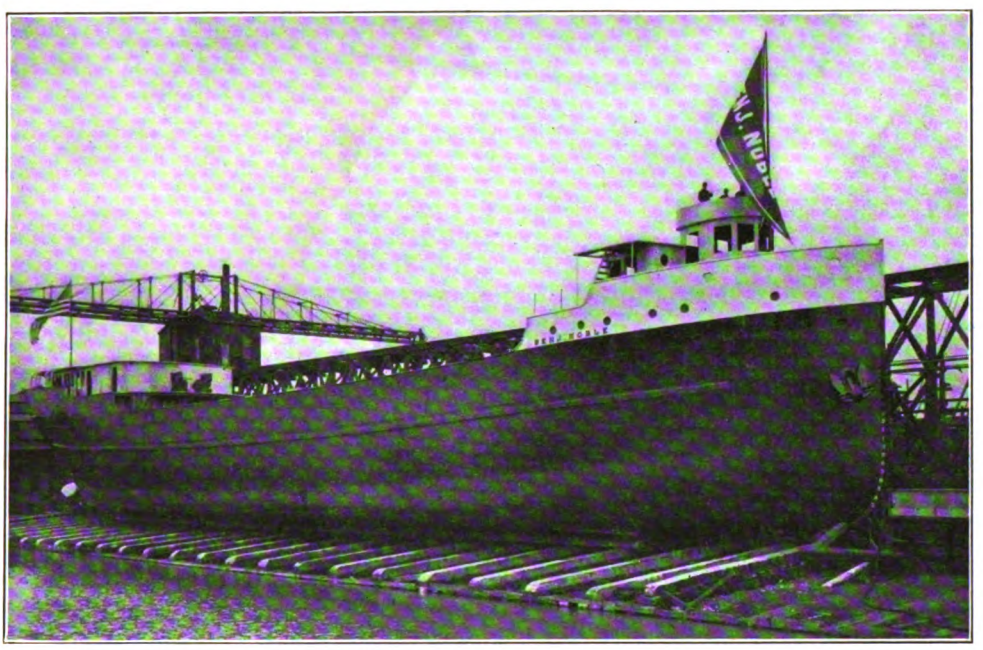
rather less than three-fourths the consumption with the three-turbine system. Generally speaking, the other figures are of the same order, although the comparison is not in every case quite so pronouncedly to the advantage of the cited combination plan.

A NEW LAKE STEAMER.

The Capital Transportation Company, of Detroit, Mich., recently launched a new ship, the *Benjamin Noble,* at the yards of the Detroit Shipbuilding Company, to enter the pulpwood trade. The ship is within a few inches of the maximum length capable of passing through the Welland Canal and locks. She is of the regular lake type, built extra strong, and is suitable for deep-sea navigation should occasion demand. The deck and hatches are built extra strong, in order to sustain the heavy deck loads which will be shipped in the pulpwood trade. Railroad iron can also be carried. The hold of the ship is one large compartment.



THE BENJAMIN NOBLE READY FOR LAUNCHING.

HULL DATA.

Length between perpendiculars....	240 feet.
Length over all.....	256 feet.
Breadth, molded.....	42 feet.
Depth, molded.....	18 feet.
Indicated horsepower.....	800
Speed	11 miles per hour.

The hull is built of mild, open-hearth steel, the sides having a tumble home of about 3 inches at the foot of the bulwarks. Between hatches the structural design is particularly heavy. The floors are spaced 36 inches apart. The bilge radius is 36 inches, and wharf protection is afforded by an oak stringer 9 by 6 inches.

PROPELLING MACHINERY.

The main engine is a three-cylinder, inverted, triple-expansion engine of 800 indicated horsepower. It is mounted on built-up girders of steel plate and angles. The low-pressure cylinder is located forward, the high pressure aft, with the intermediate between. The cylinder walls are 1⅛ inches thick. The pistons and cylinders are of cast iron. The piston rods are of steel, 4 inches in diameter, and the connecting rods of wrought iron, strap connected and babbitted. The crank shaft is built up of cast steel slabs with mild steel shafting and pins. The engine is designed to turn 90 revolutions per minute. The cylinders are 17, 24½ and 46 inches diameter by 36-inch stroke. The valves are actuated by the Stevenson link motion, and the main steam pipe is 7 inches in diameter.

The steam-producing plant is forward of the engine room. It comprises two boilers, 11 feet 6 inches long by 12 feet diameter. Natural draft is used, and the steam pressure is 180 pounds per square inch. Each boiler has two furnaces, 42 inches in diameter, and 174 3½-inch tubes. The total grate surface is 75 square feet, and the total heating surface 2,700 square feet, giving an approximate ratio of 1 to 36. The boilers are arranged athwartship in the stokehold, and are fired from the forward end. The coal is carried in bunkers alongside the boilers, the coal being distributed through a V-shape hatch over the boilers.

The *Noble* has a single cast iron screw, 12 feet in diameter and 12 feet 6 inches pitch, with four sectional blades.

AUXILIARIES.

The auxiliary machinery consists of a dynamo, direct connected in the engine room for lighting and four hoisting engines on the deck for loading cargo. The hoisting engines, ballast pump, main feed pump, steam windlass and two deck engines are all piped from the main boilers. The steering gear is the Detroit steam gear.

The crew's quarters are divided, part of the crew having quarters forward and the rest aft. The engineers' staff is housed aft, while the mates, wheelsmen, firemen and watchmen are housed forward.

The Cunard steamship *Lusitania* has again lowered the time of passage across the Atlantic Ocean, having left Liverpool Saturday evening, Aug. 28, and landed her passengers in New York before 8 o'clock on Thursday evening, Sept. 2. The time from Daunt's Rock to Ambrose Channel Lightship was 4 days 11 hours 42 minutes, and the average speed 25.85 knots. This trip enabled passengers to land on the fourth day from Queenstown.

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KRUPP SUBMARINES FOR THE AUSTRIAN NAVY.

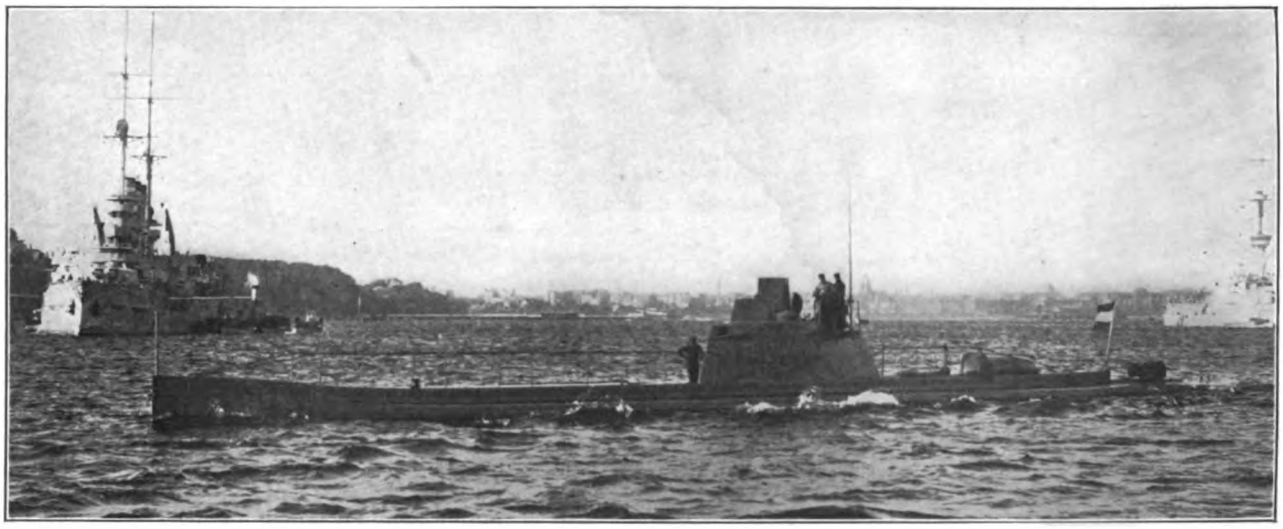
BY FRANK C. PERKINS.

Two submarine boats for the Austrian navy have recently been built at Kiel at the Germaniawerft of Fried. Krupp Aktiengesellschaft. They have an under-water displacement of 300 tons and a surface displacement of 235 tons, their extreme length being 142 feet. The surface draft is 9 feet 8 inches, while the breadth is 12 feet 4 inches outside measurement of the double hull. The inner hull is a cigar-shape, watertight shell, having a structural strength calculated to resist the water pressure at a depth of 165 feet.

The watertight hull is constructed with nine welded circular sections, three of which at each end are slightly conical, while the three sections amidships are cylindrical in form. There would allow communication with the outside atmosphere under certain conditions, and the buoy is mounted on deck, and so arranged as to be unfastened from the inside of the hull, thereby establishing connection by telephone with a rescuing crew if found necessary at any time.

These Austrian submarines are capable of making 12 knots as a surface speed, the submerged speed being 8.6 knots. They have a radius of action of 60 miles at a speed of 6 knots when submerged, the radius of action above water being 1,200 miles, at an economic speed of 10 knots.

Each boat is provided with two torpedo tubes, and carries a supply of three 18-inch Whitehead torpedoes. Strong nickel-steel plates are used in the framing of the conning tower, amidships; these plates being capable of resisting the attack of small guns. All of the equipment for controlling the boats



NEW SUBMARINES FOR THE AUSTRIAN NAVY, DESIGNED TO OBTAIN SEAGOING QUALITIES.

are several watertight compartments provided, the hull being sub-divided by means of bulkheads with the torpedo armament in the bow section, the next being occupied by the electric storage battery room and a compartment for the crew, with galley and electric cooking apparatus. The sections amidships contain the inner ballast tank, while the stearing gear for the two pairs of diving rudders is located just below the conning tower. The next section is occupied by the engine room, containing the electric motors and the internal-combustion engine; storage battery installation being provided for in the last water-tight section.

The shape of the outside hull is similar to that of an ordinary torpedo boat, a weather deck extending the entire length of the craft, which is utilized by the crew when traveling on the surface. Most of the water-ballast pipes are fitted between the inside hull and the deck platform, as well as all the kerosene pipes.

For surface navigation the boat is propelled by two 600-horsepower, two-cycle oil engines, while two electric motors are provided for propulsion when the boat is submerged, the electric motors driving two reversible screws and developing 320 horsepower. There are two main bilge pumps and one auxiliary pump, all operated by electric motors, with two hand-operated bilge pumps installed in the engine room with the air compressors and other accessories.

A most interesting feature is the 5-ton safety keel, which can be detached simply by the movement of a handle working an ordinary gear. In order to prolong the stay under water and to reduce the liability of accident to a minimum, a number of appliances have been installed for purifying the vitiated air. On the outside plating air connections are arranged which in action is installed in the conning tower, including two periscopes, the only opening being directly above the conning tower.

There is a conning platform for surface navigation arranged aft of the conning tower, the latter being enclosed in a structure designed to diminish the resistance when traveling under water, and therefore allowing ship-shape lines. It will be noted that there is a decided effort to obtain seagoing qualities in these submersibles.

MAST AND DERRICK MOUNTINGS.

LIFT AND PURCHASE BLOCKS FOR 25-TON DERRICKS.

Fig. 1 shows a treble lift and purchase block for a 25-ton derrick. The sheaves are 14 inches in diameter by 2½ inches

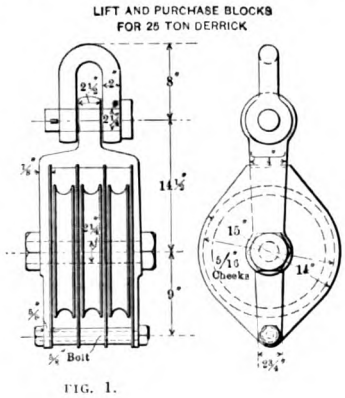


FIG. 1.

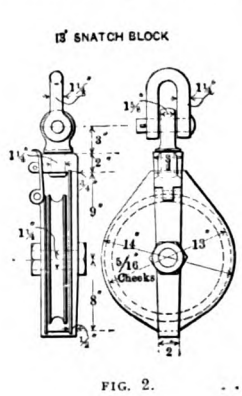


FIG. 2.

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thick. The pin is 2¼ inches in diameter, grooved for oil and fitted with a feather at the head to prevent turning. A 5/16-inch screw is fitted in the nut to prevent loss. From the center of the sheave pin to the center of the shackle pin is 14½ inches; from the center of the sheave pin to the center of the bolt through the distance piece is 9 inches.

The head takes a 2-inch shackle, and is 2½ inches thick; the pin of the shackle is 2¼ inches in diameter, and is fitted with a split forelock pin. The jaw of the block at the crown is 4 inches broad by ⅞ inch thick; at the distance piece it is 2¾ inches by ⅝ inch thick. The checks are 5/16 inch thick; the division plates are ⅛ inch thick, and the overall width of the block is 15 inches.

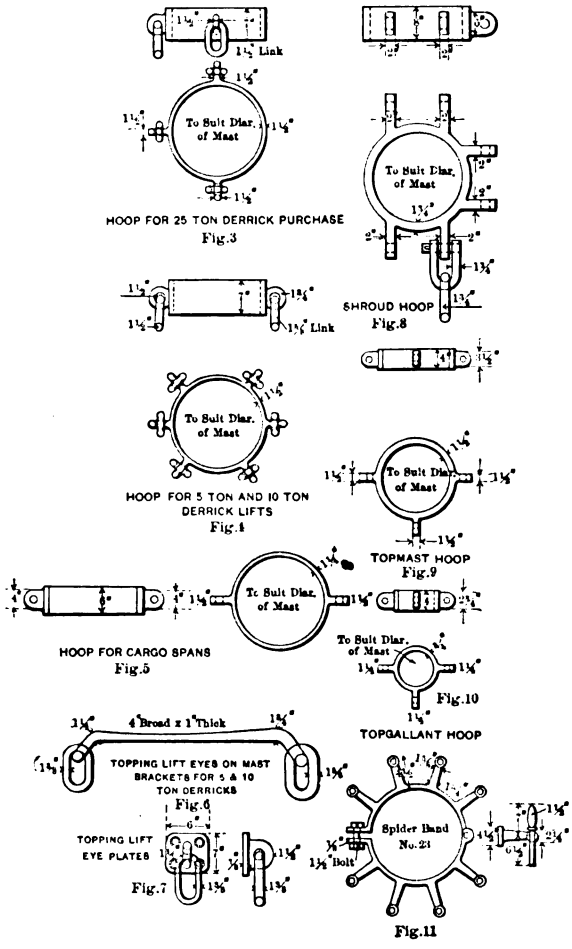
In drawing blocks for derrick arrangements, it is advisable to make inquiries as to which way the head of the derrick looks; one will be with the head as drawn, and the other with the head across the block, and two will be fitted with beckets.

A 13-INCH SNATCH BLOCK.

Fig. 2 shows the dimensions and details of construction of this block.

HOOPS FOR DERRICK LEAD BLOCKS, LIFTS, ETC.

Fig. 3 shows a hoop for taking 25-ton derrick purchase blocks. It is 7 inches deep, 1½ inches thick, and is shrunk on



to the mast; then the holes are carefully bored for rivets. Three eyes are shown in this case, 1½ inches in diameter to take 1½-inch links; to these links are shackled the lead blocks as detailed.

A hoop to take 6 derrick topping lifts, 3 on fore side of mast and 3 on after side, is shown in Fig. 4. The eyes to take 5-ton lifts are 1½ inches in diameter, with 1½-inch links to take double-purchase blocks. The eyes to take the 10-ton derrick lifts are 1¾ inches and the link is 1⅝ inches in diameter. Care should always be taken with those links, to see that shackles of the blocks ship and unship easily.

Cargo span hoops are shown in Fig. 5. The hoop is 6 inches deep, 1¼ inches thick, and two snugs are fitted to take the shackles of 4-inch S. W. R. stays. The snugs are 4 inches deep and 1½ inches thick.

Fig. 6 shows the usual method of fitting topping lift plates on mast brackets. The plate is 4 inches broad by 1 inch thick, and eyes are worked on each end. The eye for the 5-ton derrick is 1¼ inches, with 1⅜-inch link. For 10-ton derricks the eye is 1¾ inches, with link 1⅝ inches.

When each lift is fitted singly to the mast, the topping lift plates are as shown in Fig. 7. The sole of the plate is 6 inches broad by 7 inches deep and ⅞ inch thick; the eye is 1½ inches in diameter, and the link is 1⅜ inches in diameter. The sole is fastened with four 1-inch rivets. The proof test of this plate would be 28 tons.

Fig. 8 shows a shroud hoop with 4 snugs. The hoop is 8 inches deep by 1¾ inches wide; the snugs are 6 inches deep by 2 inches thick. The snugs are to take 1¾-inch shackles, and to the shackles are fitted 1¾-inch links, the links taking double 3½-inch shrouds. On the fore side of the mast are fitted two preventer stays.

A topmast hoop (Fig. 9) is made 4 inches deep, 1 inch thick, with 3 snugs, which are 3½ inches deep and 1½ inches thick, arranged to take 1-inch shackles. The stays are 3¼-inch S. W. R. Fig. 10 shows a topgallant hook, which is 4 inches deep by ¾ inch thick, with 3 snugs 2¾ inches deep and 1¼ inches thick. They are arranged to take the ¾-inch shackles of 2½-inch S. W. R. The usual hinged spider band, with 8 belaying pins, is shown in Fig. 11. It is 4 inches deep by ¾ inch thick. The detail explains itself.

A Marine Steam Turbine Reducing Gear.

Ever since the introduction of the steam turbine for ship propulsion numerous efforts have been made to improve the efficiency of turbines running at comparatively low speeds, in order to accommodate them to the most efficient propeller speeds. The result in nearly every case has been a more or less unsatisfactory compromise, since the turbine is essentially a high-speed engine, and its best efficiency and its lightest weight per horsepower developed can only be obtained at high speeds. To overcome this difficulty Rear Admiral George W. Melville, formerly chief engineer of the United States navy, has designed a reducing gear to be interposed between the turbine and the propeller shaft, so that the turbine can run at a comparativeley high speed, while the propeller shaft revolves at relatively low speed. According to the *American Machinist* the gear consists of a floating frame supporting two spiral gears of different diameters, the smaller of which is connected to the turbine shaft and the larger to the propeller shaft. An experimental gear of this type has been built at the shops of the Westinghouse Machine Company, Pittsburg, Pa., under the personal supervision of Admiral Melville and John J. Macalpine. It is designed to transmit 6,000 horsepower. The pinions are of steel, having a tensile strength of 90,000 pounds per square inch, and the gears are 22-inch face and 14 and 70 inches pitch diameter.

The pinions each have thirty-five teeth and the spur wheels 176, a hunting cog being introduced to equalize the wear. In order to secure comparatively noiseless operation a small pitch was necessary; the pitch in this case was made 1¼ inches, and the pitch helices were placed at an angle of 30 degrees with the axis of the shaft. One wheel and pinion have right-handed and the other pair left-handed helices, in order to eliminate end thrust on the shaft. The small pitch, of course, necessitated the use of broad teeth.

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THE DESIGN OF TURNING ENGINES.

BY EDWARD M. BRAGG, S. B.

When engines are being overhauled, or when the valves are being set, it is often necessary to turn the crank shaft over. In small engines this can be done by means of a bar fitting into holes drilled in one of the coupling flanges. In larger engines, developing from 500 to 1,000 indicated horsepower, a worm and wheel such as is shown in Fig. 1 can be used, and in engines over 1,000 indicated horsepower it is customary to use two worms and two worm wheels connected to a steam engine of one or two cylinders (see Fig. 2). Such an arrangement gives a large multiplication of power in a small space and enables one to use small steam cylinders.

In Fig. 1 the worm is carried in an eccentric bearing, so that by turning the bearing the worm is thrown out of gear and the worm wheel is free to revolve with the crank shaft. In Fig. 2 the large worm C is attached to the shaft by means of

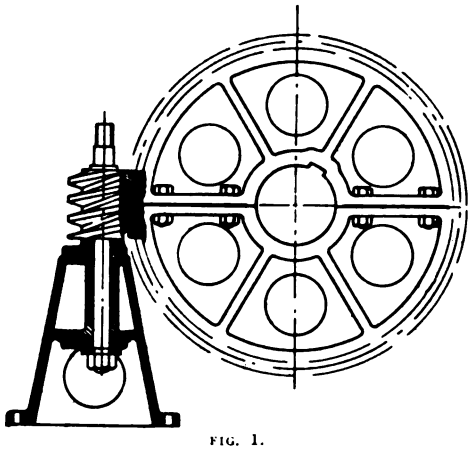

FIG. 1.

a removable key or pin, or in some other way, so that when it is not in use it can be screwed to the upper part of the shaft and the worm wheel D will be free to revolve. In some cases the shaft of the large worm is not fixed at the lower end, but is held by a rod operated by a hand wheel. The shaft has in the upper part a Hooke's joint, which permits the lower part of the shaft and the worm to be swung back out of gear.

In Fig. 2 the small worm A is keyed to the crank shaft of the turning engine, and drives the small worm wheel B. Upon the same shaft with the latter is the large worm C, engaging with the large worm wheel D, which is usually at the aft part of the engine, and when convenient is mounted upon the coupling between the crank shaft and the thrust shaft.

WORM WHEELS.

The turning engine should be of such size as to be able to turn the main engine over once in from five to ten minutes. The revolutions per minute of the engine are usually between 200 and 400, so that it will make from 1,000 to 4,000 revolutions for one turn of the main engine crank shaft. If the worm A, Fig. 2, on the turning engine crank shaft is single-threaded, the turning engine will have to make as many revolutions as there are teeth on the small worm wheel B in order to revolve it once. The same is true of the large worm C driving the large worm wheel D on the main engine crank shaft; hence the number of teeth on the two worm wheels must be such that

$$n\, n_1 = r\, m \qquad (1)$$

n = number of teeth on small worm wheel.
n_1 = number of teeth on large worm wheel.
r = revolutions per minute of turning engine.
m = number of minutes required for one revolution of the main engine crank shaft.

There is no fixed relation between the value of $r\, m$ and the indicated horsepower of the engine. But, roughly speaking, for engines of 1,000 indicated horsepower, $r\, m = 1{,}000$; for engines of 3,000 to 5,000 indicated horsepower, $r\, m = 2{,}000$; for engines of 8,000 to 10,000 indicated horsepower, $r\, m = 3{,}000$.

In order that the large worm wheel may be removed without disturbing anything else, it is usual to make it in two parts, so the number of teeth on this wheel must be even. The diameter of the wheel will be limited by the height of the crank shaft above the foundation. It will be placed as close to a bearing as possible, and it will be found that the diameter of the pitch circle of the wheel cannot be much more than 1.5 times the stroke of the main engine without making it difficult to have access to the holding-down bolts in that neighborhood. The diameter will usually be from 1.1 to 1.5 times the stroke of the main engine.

For practical reasons it is not advisable to use a pitch of less than 1.75 inches upon the small worm wheel, and the pitch is usually from 1.75 to 2.25 inches. The pitch used on the large worm wheel is usually from 2.25 to 3.5 inches.

The proportions of the teeth on the worms and wheels should be about as follows:

Length of teeth = .65 pitch.
Face of teeth = .3 pitch.
Flank of teeth = .35 pitch.
Thickness of teeth at pitch circle = .48 pitch.
Breadth of teeth at root of worm wheels = 2 to 2.5 pitch, or such that the arc of contact between worm and wheel is about 60 degrees.

Least number of teeth on small worm wheel = 25 to 30. When necessary, the thickness of the teeth of the worm wheel at the pitch circle can be made more than .48 pitch if the worm is made of a stronger material than the wheel.

The size of the teeth cannot be determined until the power which they are to transmit is known. The size of the turning engine cylinder, or cylinders, will depend upon the initial friction of the engine, the efficiency of the transmission gear, and the pressure at which the turning engine takes steam. The turning engine takes steam from the auxiliary steam line, and its pressure may be anywhere from 50 to 100 pounds per square inch.

FRICTIONAL RESISTANCE TO BE OVERCOME.

The power which the turning engine delivers to the crank shaft must be sufficient to start the engine from a state of rest. The frictional resistance to be overcome cannot be taken to be the same as the so-called initial friction determined from engine trials where the engine is run at as low speeds as possible, and the curve of initial friction is continued to the point of zero revolutions. It is well known that for most substances the coefficient of friction is larger in starting from a state of rest than when in motion. The piston speed in these tests for initial friction seldom gets below 200 to 250 feet per minute. When the turning engine is in operation, however, the main engine turns over only once in from five to ten minutes. It is probable, also, that the bearings and piston rods will not be as well oiled at this time as when the engine is under steam, and the friction of the piston rings will be greater, as they have not the lubrication that the condensed steam on the cylinder walls affords. So the initial friction must be multiplied by the ratio that the coefficient of friction from a state of rest bears to the coefficient of friction of motion, and also by a factor to allow for the worst condition of bearings and cylinder walls.

Experiments made by Professor Thurston show that with a cast iron journal in a steel bearing under a pressure of 50 pounds per square inch, and lubricated with lard oil, the coefficient of rest was .07, while when moving at a speed of 150 feet per minute the coefficient was .02. In experiments made

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by Mr. Wilfred Lewis, reported in the *Proceedings* of the A. S. M. E., vol. 7, p. 273, the coefficient of friction between the teeth of gears was found to vary, as shown in the following table:

Velocity of sliding feet per minute—	3.15	5.24	10.5	21	52.5	105	210
Coefficient of friction—	.095	.088	.074	.059	.038	.026	.020

When these are plotted, the curve gives the coefficient of friction for a velocity of 0 feet per minute as .108. This makes the coefficient of friction at 0 velocity about 4.5 times that at 150 feet per minute. We seem warranted, then, in assuming that the frictional resistance that must be overcome in an engine which is barely moving will be about four times as great as the initial friction of the engine at the piston speed at which tests are usually made. We will assume, also, that the condition of the bearings, bearing surfaces and cylinder walls at the time the turning engine is used may be such as to increase the work of the engine by 50 percent.

The average initial friction is given by Taylor as being of such an amount as to require 2 pounds mean effective pressure upon the area of the low-pressure piston to overcome it. Blechynden gives from 1.5 to 1.75 pounds mean effective pressure over the area of all the pistons as the force necessary. With the ordinary ratio of cylinder areas this gives from 2.5 to 2.75 pounds mean effective pressure referred to the low-pressure cylinder. The analysis of the *Yorktown* trial gave 1.6 pounds mean effective pressure referred to the low-pressure cylinder as the amount absorbed by initial friction; in the *Kearsarge* it was 1.95 pounds, and in the *Massachusetts* 1.5 pounds.

All things considered, it would seem wise to assume that the mean referred pressure necessary to overcome the initial friction will vary from 2 pounds in naval and other lightly-built engines, where the parts are small relative to the power developed, to 2.5 pounds in the ordinary type of triple engines for merchant ships, and up to 3 pounds for six-cylinder quadruple engines.

EFFICIENCY OF WORM GEARING.

The efficiency of worm gearing varies considerably, depending upon the care taken in its design. Of course, as much care need not be taken in the case of gears for a turning engine, which is used but seldom, and then only for a few minutes at a time, as would be necessary in the case of gears for machine tools. Nevertheless, it is well to bear in mind the principles upon which good design depends. First, the angle of the thread should be as large as possible up to 30 degrees or more for worm gearing, where the maximum efficiency is obtained. With spiral gears the maximum efficiency occurs at an angle of about 45 degrees, but the shafts are not at right angles, so there is not as much side thrust upon the spiral gears as there would be in the case of a worm and wheel with this angle of thread. In order to get these large angles it is customary to use double, triple and quadruple threads. Single threads are always used in turning gears, so the thread angle will not be greater than 9 degrees or 10 degrees, and is more often less than 7 degrees.

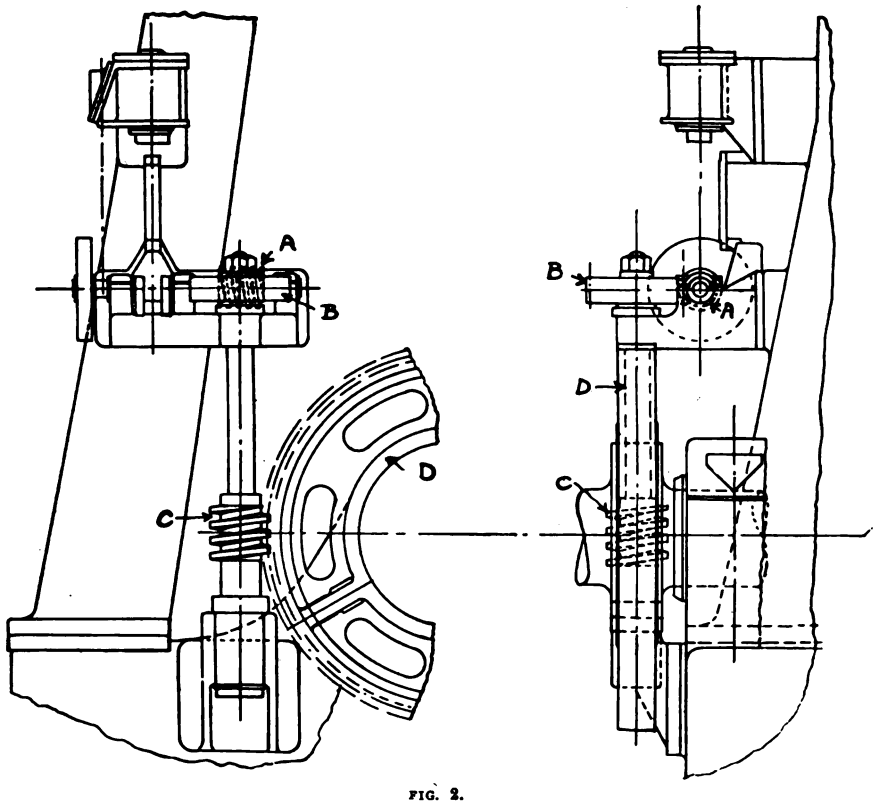

FIG. 2.

Second, the efficiency increases with the velocity at the pitch circle of the worm up to a speed of about 250 feet per minute.

Third, in order to prevent cutting and keep the worm cool, the thrust upon the teeth should decrease as the velocity at the pitch circle increases.

Fig. 3 gives the results of experiments made by Mr. Lewis for William Sellers Company. These curves seem to be in accord with the results obtained from practice (see *Worm and Spiral Gearing*, by F. A. Halsey). These curves show the advantage of using large pitch angles. The efficiencies shown by Fig. 3 can be obtained if the worm is run in an oil bath, and if some care is taken in the design of the thrust bearing for the worm.

Fig. 4 shows the results of tests by Prof. Thurston upon three worms. In the one marked "collar," the thrust was taken by the end of the shaft; in the one marked "button," it was taken by the point of a set screw; in the one marked "roller," it was taken by ball bearings. The lower curve represents the conditions that one is likely to meet in the case of worms for

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turning gears, so it would appear that an efficiency of .4 is about all that will usually be obtained. The lost work, .6 of the total work, will be used up in overcoming friction between the teeth, friction in the thrust bearing of the worm, and friction in the bearing of the worm wheel due to the force acting upon the teeth, at the pitch circle and to the side thrust of the worm. We will assume that the loss in each of the above places mentioned is .2 of the total power driving the worm. In figuring the teeth, then, we shall assume that .6 of the power driving the worm passes through them.

Experiments have been made to determine the relation between the speed at the pitch line of the worm and the force acting normal to the face of the teeth when cutting begins.

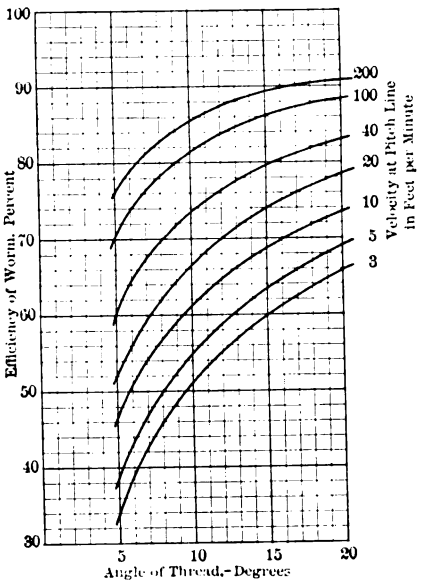

FIG. 3.

Lewis found that with a velocity of sliding of 306 feet per minute, a force of 5,558 pounds produced no cutting and no change in efficiency. With a speed of 360 feet per minute and a force of 4,837 pounds, cutting commenced after running six minutes. With a velocity of 400 feet per minute and a force of 3,481 pounds, cutting commenced after three minutes. The products of these quantities are as follows:

Velocity of sliding, feet per minute	306	360	400
Force × velocity	1,700,748	1,741,320	1,392,400

The experiments made by C. Bach and E. Roser, republished in the *American Machinist*, July 16 and 23, 1903, show smaller values of "force × velocity," but the curves given show that the value of this product increases as the velocity decreases.

The mechanical efficiency of the turning engine will probably be not less than .8, and if the cut-off in the cylinder is about .7 of the stroke the mean effective pressure will be about .75 of the initial pressure absolute. The turning engine takes steam from the auxiliary steam line, where the pressure will probably not be more than 100 pounds, and very often it will be less than that. It is advisable to put in a large enough cylinder so that the engine will run with an initial pressure in the cylinder of 50 pounds gage, or 65 pounds absolute. The stroke of the turning engine is usually from 6 to 8 inches, and the diameter also is usually within this range. When the conditions are such as to call for more than 8 inches diameter, it is advisable to use two cylinders.

Fig. 3 shows that the efficiency of worm gearing increases as the pitch angle becomes larger. In order that the pitch angle of the worm may be as large as possible, it is best to keep the diameter as small as possible. If the small worm is

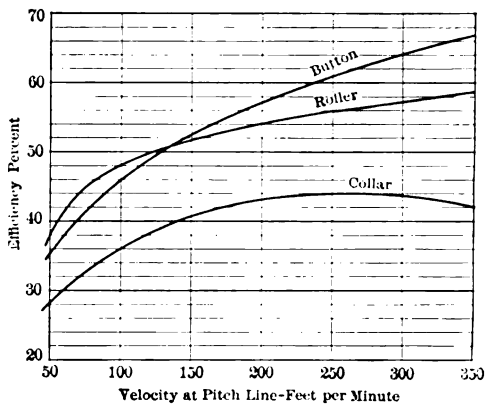

FIG. 4.

cut out of the turning engine crank shaft forging, the diameter of its pitch cylinder can be about 2.5 times the pitch of the teeth. If it is a separate piece keyed to the shaft, the pitch cylinder will have a diameter of about 3 times the pitch of the teeth, in order that the thickness of metal below the root of the thread may be from .5 to .6 times the pitch of the teeth. The length of the worm should be from 3 to 4 times the pitch of the teeth.

If all of our assumptions are correct, we find that the frictional work to be overcome in one revolution of the engine is

$$\frac{\pi D^2}{4} \times MEP_f \times f \times 2S \times g \qquad (2)$$

Where D = diameter of low-pressure cylinder of main engine in inches.

MEP_f = frictional mean effective pressure referred to the low-pressure cylinder.

= 2 for naval and other light engines.

= 2.5 for engines of merchant ships.

= 3 for six-cylinder quadruple engines.

f = ratio between coefficients of friction of rest and motion = 4.

S = stroke of main engine in inches.

g = coefficient to allow for worst condition of bearings and surfaces = 1.5.

(To be continued.)

Combination Reciprocating Engine and Turbine-Driven Ships Show High Efficiency.

The White Star liner *Laurentic*, built by Harland & Wolff, Belfast, for the Canadian trade, is the second large steamship equipped with a combination of reciprocating engines and turbines for propulsion. She is propelled by three screws, the two outboard screws being driven by four-cylinder, triple-expansion engines, and the center screw by a single low-pressure Parsons turbine. Steam is supplied at a pressure of 200 pounds per square inch, and is passed first through the reciprocating engines, where it is expanded down to a pressure of from 14 to 17 pounds per square inch absolute; then it is used in the low-pressure turbine, being exhausted finally into the condenser, where a very low vacuum is maintained. The propelling machinery was designed to develop 10,000 horsepower, to give the ship a speed of 15 knots. On the trial trip, however, 12,000 horsepower was developed, and a speed of 17½ knots attained. It is reported that the coal consumption per indicated horsepower per hour was 1.1 pounds, and that the steam consumption per indicated horsepower per hour was 11 pounds. These results bear out the claims for economy made for this system of propulsion by the builders of the New Zealand liner *Otaki*.

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DUTCH MARINE SUCTION-GAS PLANTS.

BY F. MULLER VAN BRAKEL.

For the small installations to which the use of producer gas is limited to-day, it is of the first importance that they can be operated with non-professional attendance. The skipper of a motor barge, often a man who formerly owned a sailing vessel and consequently not at all familiar with engines, should be able to manage the whole plant himself. He should remain at the rudder when under way, leaving the firing and oiling to a boy or a deck hand, and only when in harbor should he have to look after the engine, clean it, and keep it in good condition. A power plant which works nicely and economically with an experienced engineer in charge may be uneconomical for the owner of the ship because of the salary of that engineer. On the other hand, with non-professional attendance it may give no end of trouble and even be less economical. Experiments made by technical men are therefore not sufficient in this case, however valuable they may be for the operator of a generator plant and motor.

To judge of the advisability of placing a producer-gas installation in a small vessel, the experience of non-professional skippers is of great, or rather of the first importance. Messrs. E. J. Smit & Son, Hoogezand, Groningen, Netherlands, have built since 1902 a good many ships with suction-gas plants, for different owners, and with many of these they have remained in touch, for varying reasons. Some owners called to have different troubles removed or to be told the unnecessary sources of them; others, when passing on their regular trips, to tell of their satisfaction, mentioning the profits made and the actual working costs. These, too, occasionally told about troubles experienced, which had been removed by skippers themselves after some seeking and trying, or which disappeared in an unexplained way—as engine troubles sometimes do when the engine is handled by inexperienced men. It was this information that made it possible to produce an engine that could be managed by non-technical men. The construction and management, the troubles experienced, the necessary room, and the weights of this installation are described in the following, while a discussion of the costs as compared with steam and oil engines will follow in a later issue.

DESCRIPTION OF THE PRODUCER-GAS INSTALLATION.

The principal parts of the producer installation are shown in Fig. 1. *M* is the cast iron producer, which for bigger installations is made of ¼-inch plate. *B* is the ash catcher, *S* the scrubber, *D* the last cleaner, filled with wood shavings. Air and steam are introduced under the producer grate *c*, and are pulled by the suction of the motor through the layers of burning coal. There the coal is first burned to carbonic acid, but higher up in the producer it is mostly reduced to carbonic oxide. The necessary oxygen is furnished by the air and by the steam, leaving the hydrogen as a desirable element of the gas mixture. The air enters through the regulating cock *R*; the water through the valve *v* into the water jacket of the ash catcher *B*, where it is heated. The water level in the jacket can never be higher than the opening of the overflow *g*, from

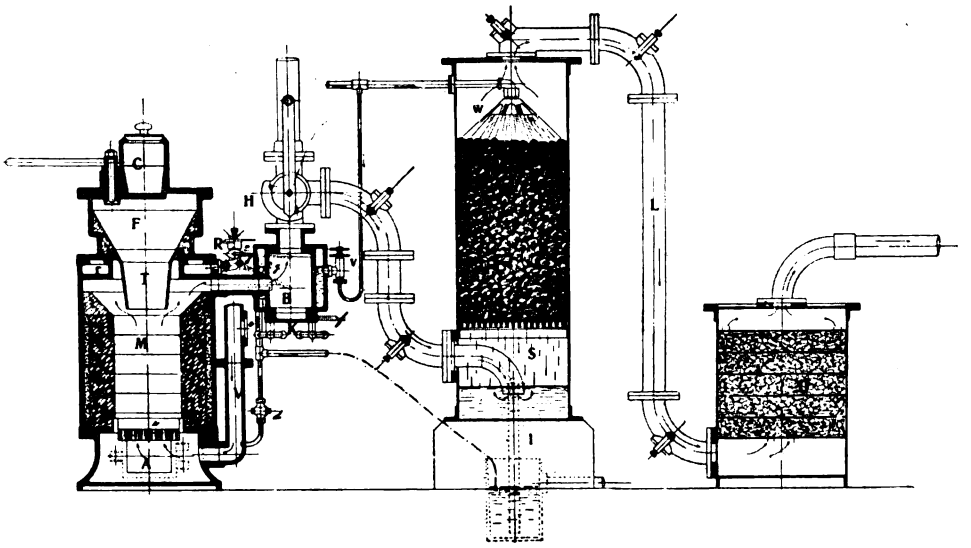

FIG. 1.—DIAGRAM OF DUTCH MARINE SUCTION-GAS PLANTS.

which the water is seen dripping into the funnel shown in dotted lines. This funnel leads to the water vessel *l*; part of the water may be led to the ashpit through cock for wetting the ashes or to bring more hydrogen into the gas mixture. From the jacket of *B* the water passes through the small tube *C* (where a cock allows regulation) to the body of cock *R*. Steam that might be formed in the jacket can proceed to *R* through the small tube *a*. In the body of cock *R* a sight-hole allows observation of the quantity of water that is seen as small jets, corresponding to the intermittent suction-action

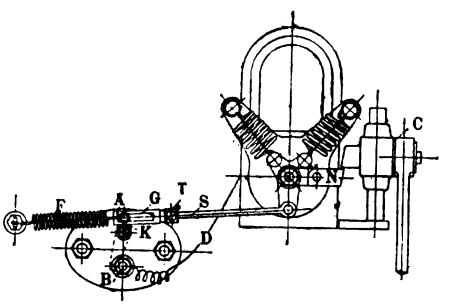

FIG. 2.

of the motor. In the ring *r* the water is evaporated and passes with the air through the down-tube *V* to the ashpit.

Thus the principal points to be observed when the engine is just started are:

1. The water dripping from *g* into the funnel—to be regulated by the water valve *v*.

2. The small water jets seen through the sight hole in air cock *R*—to be regulated by the cock in tube *C*, and by the air cock itself. For diminishing the opening for air means a stronger pull exerted on the water.

3. If steam is seen coming out of *g*, the cock in tube *a* must be opened, to lead the steam formed in the water jacket to the air cock and evaporator ring.

From the ash catcher the gas passes through a 3-way cock *H*, which leads to the atmosphere or to the scrubber-bottom.

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Fig. 3 shows another arrangement of the piping between ash catcher and scrubber, which has advantages. The end of the gas pipe dips about ⅞ inch into the water under the scrubber, in this way preventing the gas from return from scrubber to producer. The water level under the scrubber is kept constant by an overflow leading to the water vessel *l*. The vertical

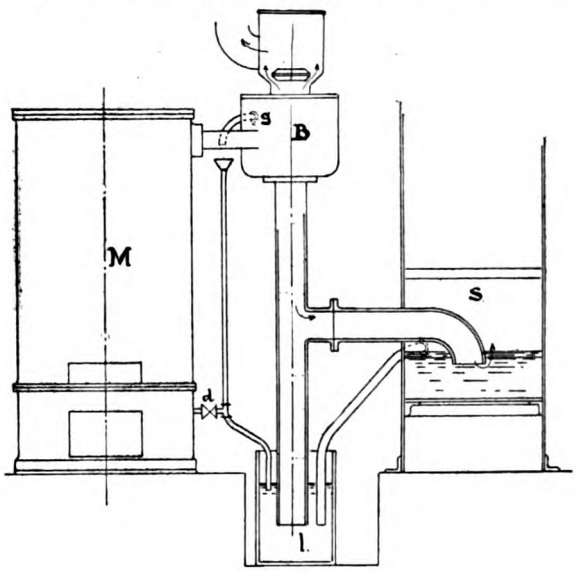

FIG. 3.

distance between the water levels in scrubber and water vessel should be at least 14 inches. When it is smaller the water in the water vessel might be sucked up into the scrubber, thereby augmenting the water resistance. The scrubber is filled with coke in pieces of about 2¼ inches by 4 inches; while water is constantly dripping through the whole column.

The last cleaner consists of a plate cylinder with six thin circular plates in it, between which the wood shavings are

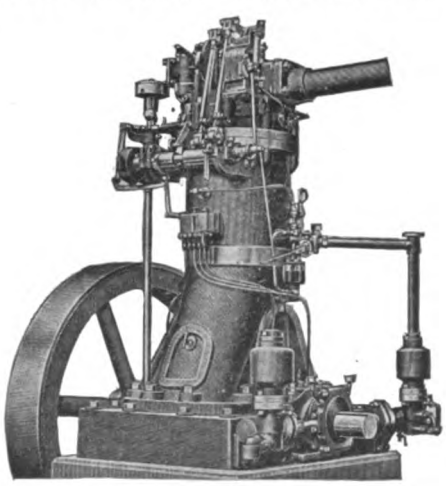

FIG. 4.

kept. The condensation water should be left off every night.

THE MOTOR.

The gas mixture on leaving the last cleaner is composed as follows:

CO	28 percent.
H	7 percent.
CO_2	5 percent.
CH_4	2 percent.
N	58 percent.

It will not burn without the addition of air. This is added in a mixing vessel attached to the motor, in the proportion of 1:1.

The motor, Fig. 4, is a vertical four-cycle engine, mounted on a low under-frame. The *A* frame is cast in one piece with the cylinder, in which a liner is fitted. The legs of the frame have doors for getting at the crank shaft and connecting rod. In the separate cylinder head the inlet and outlet valves are mounted vertically, both of them kept shut by springs and opened by levers driven by cams on a horizontal cam shaft, which makes half the revolutions of the crank shaft. The regulator works a throttle valve, regulating the passage for the air and gas mixture.* This gives at all speeds well-formed diagrams, though differing in size (Fig. 5).

The ignition is electro-magnetic, details of which are shown in Figs. 2 and 6. The spark-drawer, *A K B*, when at rest, touches the end of the pin *P*, which takes the current from *D*.

FIG. 5.

The part *C*, moving downward, takes *N* with it, thereby pushing the rods to the left, leaving *A* free, however, because it slides in slot *G*. When *C* releases *N*, rod *S* springs back to the right, taking *A* with it for a very short moment, thus breaking the current at *B* and giving the spark.

On the forward end of the crank shaft an eccentric is fitted, driving a cooling-water pump and a bilge pump, mounted opposite each other. The after end of the crank shaft bears the flywheel with the clutch inside.

STARTING.

For starting a hand-operated fan is used, placed on the tube *V* at *c*. It blows the outer air under the grate, through the producer, ash catcher and 3-way cock *H* to the atmosphere. The producer is kindled with wood and a few loads of coal, which are put into the producer through the lock *C*. While

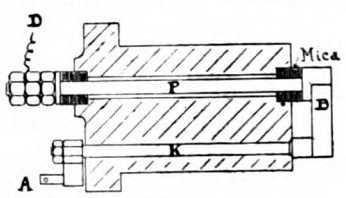

FIG. 6.

the fan blows the fire the air-and-water cocks remain shut and the 3-way cock *H* leads to the atmosphere. After some ten minutes more coal is locked into the producer till it is filled to the under side of the funnel *T*. The fan is kept blowing till at last the greater part of the coal is glowing well, then two more loads of coal are put on and the 3-way cock turned to the motor. Somewhere in the piping between the last cleaner and the motor a gas-trying cock is fitted, and here the gas is now tried. It should burn with a dull orange flame, that is not easily blown out. The motor is started by turning the flywheel by hand and then the fan is stopped. The producer now only gets air through the body of the fan, but the air-and-water cocks are opened as soon as the motor is going well, the valve between fan and air pipe *V* being shut next. After some thirty minutes the steam cock in tube *a* is opened.

After stopping for some time the motor is started in the same way. When the gas is tried it burns with a blue flame, showing the presence of hydrogen in the mixture.

PERIODICAL CLEANINGS.

The inside of the producer should be cleaned twice a week. The grate every two hours. This is done through the bottom

* Fig. 4 shows an old form of regulator regulating the quantity of gas only.

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door after wetting the ashes and poking through the upper door.

The scrubber coke should be renewed once a year. It may then be burned in the producer, mixed up in the anthracite in the proportion of 1:4, after being ground to small pieces.

The wood shavings in the last cleaner must be renewed twice yearly, the piping being cleaned at the same time.

TROUBLES.

Motors that never give any trouble are to be found only in advertisements—in fact, all those motors have that desirable quality. In practice, on the contrary, troubles are rather frequent, as most motor men will know. It should be acknowledged, however, that most of the troubles are not so much the fault of the motor, or its manufacturer, as of the man who handles it. Motors that are well beyond the experimental stage never need to trouble their owners. But here comes in the difference between a sensitive and an easy plant. The steam engine is an example of an easy plant. Many things may be said against it, but it *always goes* when the steam valve is opened. On the other side stands the suction-gas installation as a typical sensitive plant. It is sometimes claimed for marine gas plants that: "Any man who can take proper care of an internal combustion engine, can, without any difficulty whatever, manage the producer-gas plant." (International Marine Engineering, Volume XIV, page 313.) This may be so, at any rate it will seem so, but it cannot be said of the plant described here. There are too many points where something may go amiss; and a man to whom the installation is new cannot at once master the whole plant. But it can be claimed that *every* man, those who never before handled an engine included, can *learn* the management in a short time. This has been proved in many instances. During that time, however, there will be troubles—with the usual bad temper and bad names for the maker. They all make themselves known by the stopping of the motor. The sources of trouble may be divided into two kinds: those originating in the producer plant; those originating in the motor itself. Of the former the following are the most common:

1. Insufficient Gas when Starting.—Some hasty skippers only fill part of the producer when starting, blow the fire through till it gives good gas and then immediately start the motor. As a consequence, the motor stops after a few minutes for want of gas. The same thing may happen after stopping an hour or more. It is then often tried to keep the motor going by filling the producer to the funnel, but this has no effect then; it is necessary first to clean the grate and then blow the fire by means of the fan till it is blowing well again. And only then, and not before, the producer may be filled.

2. Bad Quality of Gas.—There may be lack of hydrogen, by bad regulating or by some water pipe being stopped up; or there may be an excess of air, by leaks in the piping or by cracks in the stone covering of the producer. Or, when encountering heavy weather at sea, the water vessel *l* may be emptied by the rolling of the ship, thereby admitting air under the scrubber. This can be avoided by a piece of wood floating on the water or by a cover over the water vessel.

3. No Gas while Working.—When there is a good fire the lack of gas always originates in too much piping resistance. Either the pipe under the scrubber dips too deeply into the water, or the wood shavings in the last cleaner are packed too stiffly. The first is caused by a rising of the water level under the scrubber, for which there may be three causes:

a. The producer coal bakes, which augments the resistance for air and steam. As a consequence there is less pressure under the scrubber and the water from the water vessel is pushed up by the atmosphere.

b. The water vessel *l* is placed too high.

c. The water level in the water vessel is too high.

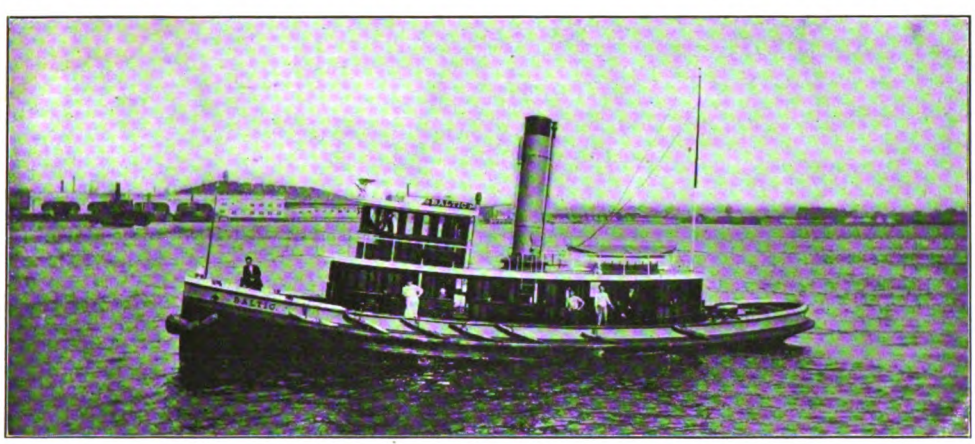
The most common troubles originating in the motor are as follows:

1. Lack of Compression.—This is to be tried by turning the flywheel by hand and observing the resistance during the compression stroke; the causes may be: leaks in the packing between the cylinder and cylinder head, leaks of the valves, leaks of the piston rings.

2. Lack of Ignition.—This is mostly caused by water entering the cylinder. Sometimes cooling water is admitted through the packing between cylinder and cylinder head. But mostly the insulation of pin *P* (Fig. 6) is disturbed by moisture, tar or dirt settling on or between the mica rings.

The ignition can be tried in two ways: by feeling it with the finger on *P*, or by looking at it through a sight hole.

(*To be continued.*)



HARBOR TUG BALTIC.

INDICATOR DIAGRAMS FROM A HARBOR TUG

BY CHARLES S. LINCH.

Indicator diagrams taken from single cylinder engines of harbor tug boats are very scarce and the writer believes that the diagrams shown herewith are the first ever taken from a Neafie & Levy tug of this size. The engine is a single-cylinder non-condensing vertical engine, having a diameter of 16 inches and a stroke of 16 inches. The clearance at the top of the cylinder is 11/16 inch and at the bottom 13/16 inch. The lead of the main valve at the top is 1/16 inch ahead and 1/8 inch astern; at the bottom it is 1/8 inch ahead and 3/16 inch astern.

Recently the writer was engaged in designing a new propeller for the tug, and, as a new cylinder was fitted on the engine at the same time, it was drilled and tapped for indicator pipes in order that accurate data regarding the performance

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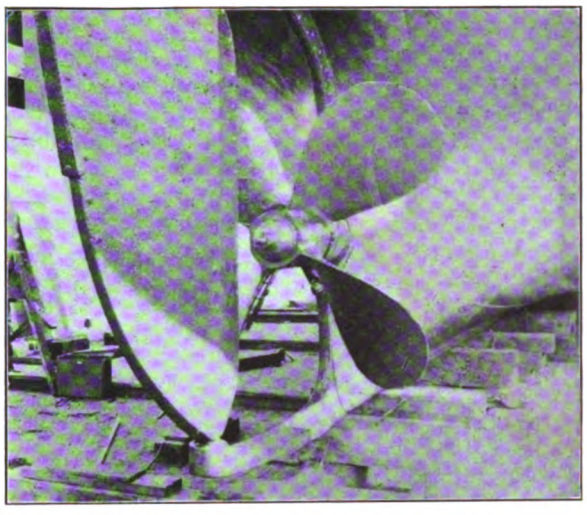
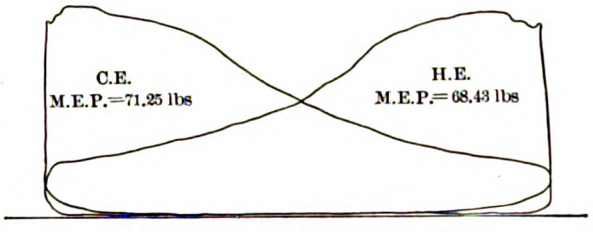


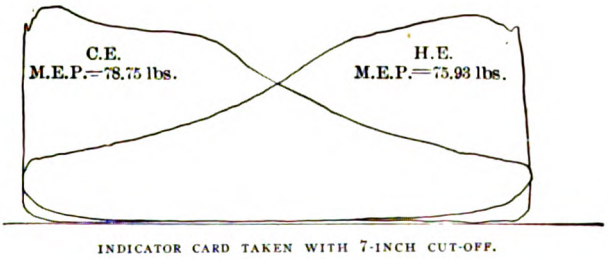
NEW PROPELLER FITTED ON THE BALTIC.

of the new propeller could be obtained. It was desired in making the changes to secure a propeller which would be a more efficient towing wheel and also a more efficient backing wheel. The revolutions were to be reduced from 165 turns when running free to not more than 135 when running free at a steam pressure of 120 pounds per square-inch gage. When cutting off at 7 inches, the revolutions at 120 pounds of steam were to be about 120.

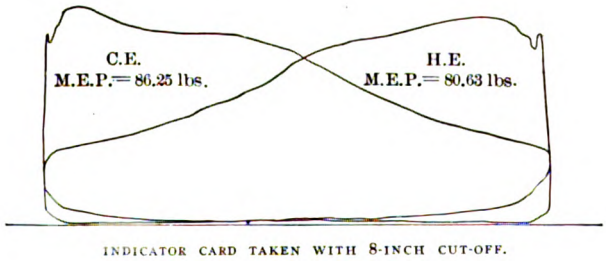
When the new propeller was designed, the power was of necessity computed, since no indicator diagrams were available. After the wheel had been fitted, and the new cylinder installed, a series of indicator cards was taken throughout the entire range of cut-off. It will be remembered that this type of engine is fitted with a Meyer cut-off valve. A set of these indicator diagrams at various cut-offs are shown herewith, together with given and computed data. These diagrams are worthy of close study, and it will be noted that the same characteristics are apparent throughout the whole range of cut-off.



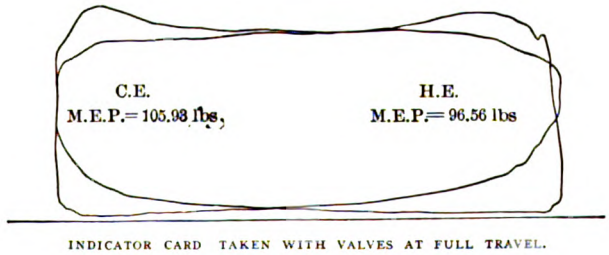
INDICATOR CARD TAKEN WITH 6-INCH CUT-OFF.



INDICATOR CARD TAKEN WITH 7-INCH CUT-OFF.



INDICATOR CARD TAKEN WITH 8-INCH CUT-OFF.



INDICATOR CARD TAKEN WITH VALVES AT FULL TRAVEL.

GRAPHICAL SOLUTION OF THRUST-BEARING PROBLEMS.

BY GEORGE E. BARRETT.

The tendency of the engineers of to-day is towards the adoption of the graphical solution of standard problems, which is due to the rapidity and accuracy of obtaining the results; this is particularly prominent where the formula has more than three variables. Generally, standard formulas are so written as to require a certain amount of rearranging in order to solve for some unknown part, but when once this equation has been plotted any one of its variables can easily be determined for any particular case.

In Fig. 1 the following variables, horsepower, knots, pressure per square inch, number, and outside diameter of the thrust bearing of horseshoe bearings and their centers of gravity, have been plotted and so arranged as to permit of easy solution of any one variable with the others being known or assumed. The general formula

$$A = \frac{217\ H.\ P.}{K\ p\ N}$$

was used, in which

A = area of one horseshoe bearing.
K = the speed of the ship in knots.
p = the allowable pressure on the bearings per square inch of surface.
N = the number of horseshoe bearings.

As can be seen, this formula was deduced from Seaton's well-known expression for the mean thrust on the thrust bearings, which, if divided by pN will give the area of one bearing.

The range of p from 20 to 60 pounds per square inch in increments of 5 pounds should be sufficient for practical cases, though finer readings can be easily obtained within 1 or 2 pounds. This applies equally as well to the curve for knots and inside diameters of the horseshoe bearings.

The diameter curves were plotted from the following formula:

$$A = a^2\ r^2\ \pi - \left\{ \frac{r^2\ \pi}{2} + \frac{a^2\ r^2\ \pi\ 2\ \theta}{360} + r^2\ \sqrt{a^2 - 1} \right\}$$

which can be reduced to the following simpler expression:

$$A = r^2 \left\{ \pi \left[a^2 \left(1 - \frac{\theta}{180} \right) - \frac{1}{2} \right] - \sqrt{a^2 - 1} \right\}$$

In which

A = the area of the horseshoe bearing.
a = the ratio of the outside to the inside diameter of the bearing.
θ = the angle as shown in Fig. 2.

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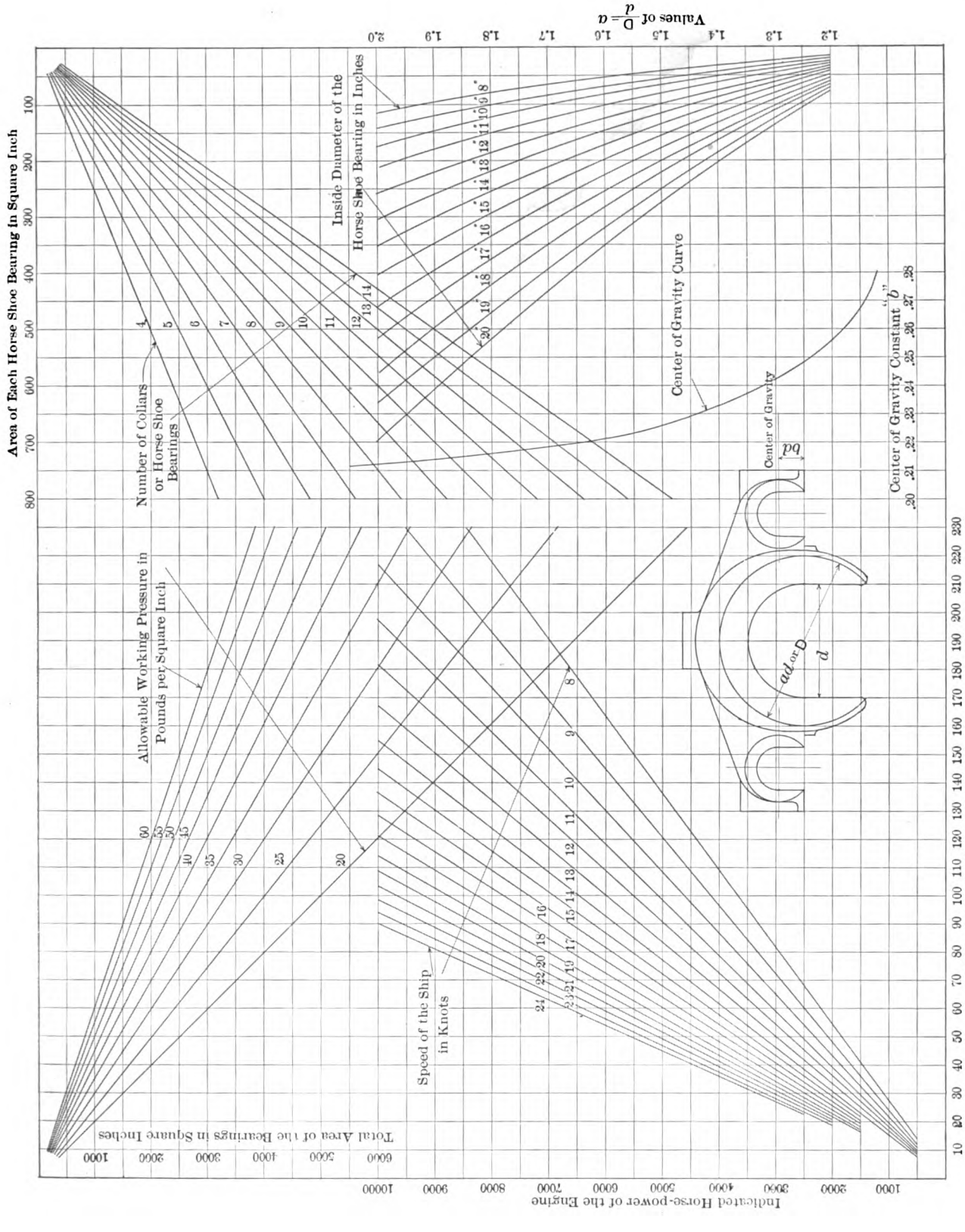
Area of Each Horse Shoe Bearing in Square Inch
Number of Collars or Horse Shoe Bearings
Allowable Working Pressure in Pounds per Square Inch
Inside Diameter of the Horse Shoe Bearing in Inches
Values of $a = \frac{D}{d}$
Center of Gravity Curve
Center of Gravity
Center of Gravity Constant "b"
Speed of the Ship in Knots
Total Area of the Bearings in Square Inches
Indicated Horse-power of the Engine
Mean Thrust in Thousand Pounds
bd
pq
ad or D
d

FIG. 1.

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The center of gravity of the bearings was computed from the formula

$$C.\ g. = \frac{r^3\ (a^2 - 1)}{A},$$

which was deduced as follows:

The center of gravity of the sector $A\ O\ B$ from the center line $X\ Y$ is $\frac{4\ R\ r}{3 \times \text{arc}\ A\ B}$. Arc $A\ B = \frac{R\ \pi\ \theta}{90}$. The moment of the sector about the center line $X\ Y$ is:

$$\frac{R^2\ \pi\ \theta}{180} \times \frac{4\ R\ r}{\frac{3\ R\ \pi\ \theta}{90}} = \frac{4\ R^2\ \pi}{6}.$$

The moment of the triangles $A\ D\ O$ and $B\ O\ C$ about the center line $X\ Y$ equals

$$\frac{1}{3} r \sqrt{a^2 - 1} \times r^2 \sqrt{a^2 - 1} = \frac{1}{3} r^3\ (a^2 - 1).$$

The moment of the semi-circular section $D\ E\ C$ equals

$$\frac{1}{2} r^2\ \pi \times .4244\ r = .2122\ r^3\ \pi.$$

The center of gravity of the horseshoe bearing will be

$$C.\ g. = \frac{.2122\ a^3\ r^3\ \pi - .2122\ r^3\ \pi - .2122\ a^3\ r^3\ \pi + a^3\ r^3 - {}^1/_3\ r^3}{\text{Area of the shoe}}$$

$$= \frac{r\ (a^2\ r^2 - r^2)}{A} = \frac{r^3\ (a^2 - 1)}{A}$$

This center of gravity formula was further arranged so that multipliers, which have been termed b, of the inside diameter of the horseshoe bearing could be easily obtained for the different values of a. This can be readily proved if desired by assuming fixed values of a in the area and center of gravity formulas, which will reduce the center of gravity formula to the simple expression bd, b being a constant for all bearings having the assumed value of a for the ratio of the inside to the outside diameter.

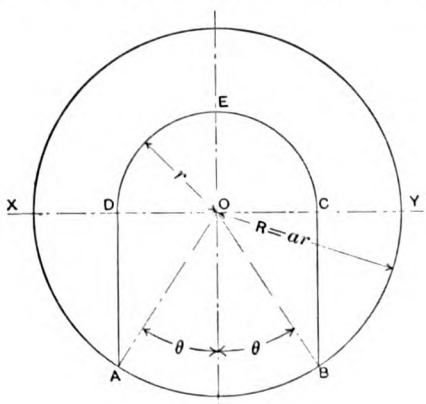

FIG. 2.

The curves in Fig. 1 have been plotted so that the mean thrust for any horsepower up to 10,000 and knots from 8 to 22 can be read; also the area of each bearing and the total area of all the bearings can be obtained.

The following examples will be used to explain the practical uses of the curves. What should be the outside diameter of the horseshoe bearings for the following conditions? Indicated horsepower, 5,000; knots, 16; allowable pressure per square inch, 35 pounds; 10 collars; inside diameter of the horseshoe bearings, 17 inches. Enter the scale for the horsepower at 5,000, and run horizontally over to the curve marked 17 for the speed in knots, then vertically upward to the curve of pressures marked 35, and over horizontally to the number of bearings curve marked 10, down vertically to the inside diameter curve marked 17, then over horizontally to the a scale. Answer, $17 \times 1.47 = 25$ inches diameter. The center of gravity of this bearing will be $17 \times .228 = 3.88$ inches above the center line. The mean thrust is 64,000 pounds and the area of each horseshoe bearing is 185 square inches.

How many collars should a thrust shaft have for the following conditions? Two thousand five hundred horsepower; 14 knots; about 30 pounds working pressure; inside diameter of the horseshoe bearing 15 inches; $a = 1.65$. Enter the scale for the horsepower and continue as explained above until the curve for the number of collars is reached. Enter the a scale at 1.65, and run over horizontally to the diameter curve marked 15, then up until the line from the pressure curve is intersected, which locates a point that falls on the curve for 6 collars. The center of gravity of this bearing will be $15 \times .22 = 3.3$ inches; above the center line, say, 3 5/16 inches.

What is the area of a horseshoe bearing that has an inside diameter 16 inches and an outside diameter 24¼ inches? $a = \frac{24\frac{1}{4}}{16} = 1.52$. Enter the a scale at 1.52, then over horizontally to the 16-inch diameter curve and up vertically to the area scale. Answer, 185 square inches.

A REMARKABLE SALVAGE JOB.

On May 13 the barge *S. O. Co. No. 91*, loaded with a full load of fuel oil, in tow of steamship *Maverick*, struck the bar at the mouth of the Columbia River. A heavy surf was running at the time, which broke the forward port sea chest and also the oil suction between the port oil pump and tanks, allowing both sea water and oil to run into the pump room,

FIG. 1.—LOOKING AFT ON THE WRECKED BARGE

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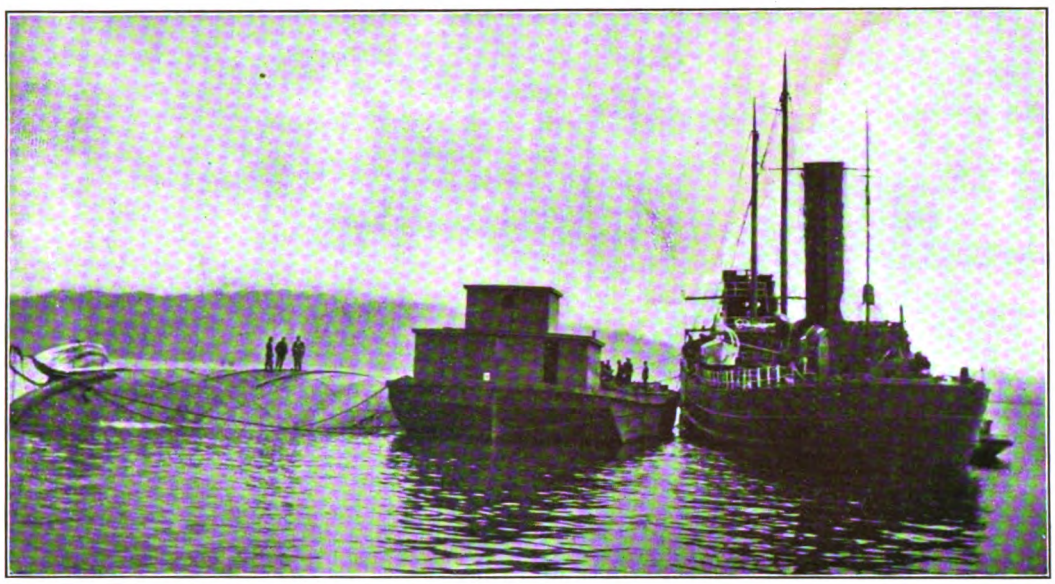

FIG. 2.—BARGE BEFORE PUMPING OUT THE OIL AND WATER.

filling this large compartment and causing the barge to sink forward and roll over on her port side in about 50 feet of water. The barge was 256 feet long with a beam of 42 feet and a depth of 25 feet.

The general opinion was that she could not be saved, but salvage operations were undertaken by Mr. D. E. Ford, marine superintendent of the Standard Oil Company. As there was from 6 to 8 feet of water over the side of the barge at low tide, it was necessary to build a cofferdam of sufficient size to allow a man with a pneumatic hammer to cut holes in the side of the barge in order to fit 6-inch suctions for pumping out the cargo in the starboard compartments. After this work was completed, and the pumps connected up and sufficient oil pumped out, the barge was moved in towards the beach, where there was about 30 feet of water at low tide.

After the barge was hauled in towards the beach a 12-inch centrifugal pump was fitted in the pump room, and when the water and oil were lowered sufficiently in this space temporary connections were made with the cargo pumps, and the oil in the port compartments was pumped into lighters and tank steamers, enabling the righting of the barge to an angle of about 30 degrees, and also permitting the placing of a lighter on the port side of the barge with suction pipes into the port tanks, in order to pump out the balance of the cargo.

Two tripods, one forward and one aft, built of heavy channels and I-beams, were rigged on the starboard side of the barge, and by means of 100-ton tackles the barge was righted on an even keel as the oil was being pumped out of the port tanks.

One of the great difficulties experienced in salving the barge was the fact that the wreckers had to contend with a heavy freshet in the river, which was constantly cutting the sand out from under the barge and allowing her to settle deeper in the sand at the rate of about 1 foot per day. Although the work of salving the barge was planned and carried out by Mr. Ford, a great deal of credit for the success of the undertaking is due to Capt. Bunting and Mr. Hague, both connected with the Standard Oil Company, San Francisco, Cal.

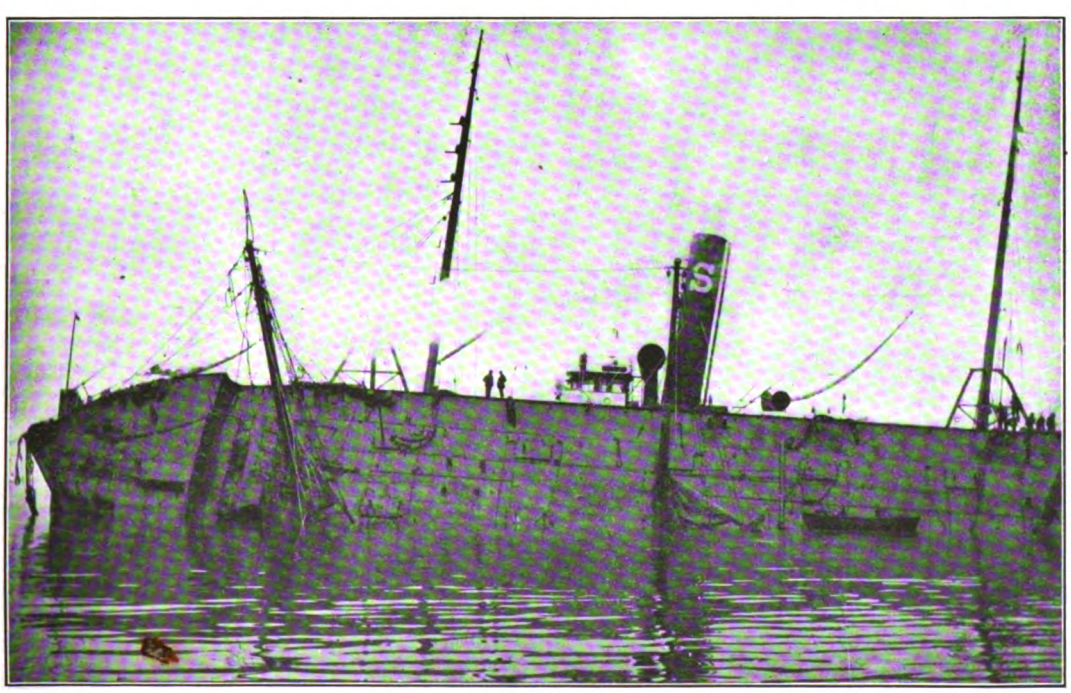

FIG. 3.—BARGE PARTIALLY RIGHTED.

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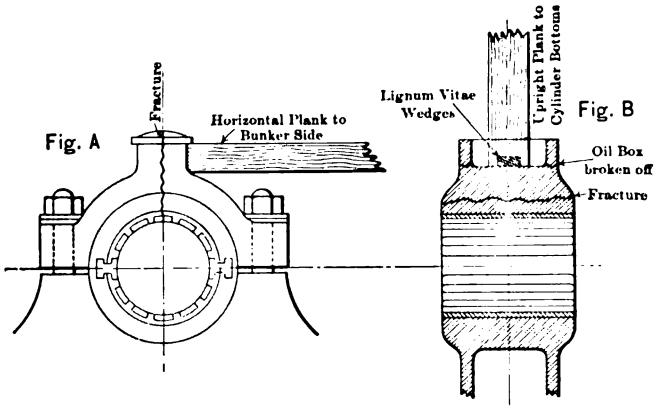
PRACTICAL LETTERS FROM MARINE ENGINEERS.*

Experiences Relating to the Design, Care and Handling of Marine Engines, Boilers and Auxiliaries; Breakdowns at Sea and Repairs.

A Critical Forty Minutes.

The experiences of a sea-going engineer very often include times of sudden stress in which every effort of concentrated resource must be exercised in order to advert disaster. An instance of this occurred some time ago in a steamer bound from Galveston to Havre with a large cargo of cotton. The ship was caught in a heavy gale with a high following sea, and those engineers who have experienced a westerly gale will hardly need to be told that a boat under such conditions will run along smoothly and steadily until an unusually large



SKETCH SHOWING FRACTURES IN MAIN BEARING CAPS AND METHOD OF HOLDING THEM IN PLACE.

roller comes along. The boat gets caught on the crest of this wave, and the result is a swept deck or something considerably more serious.

Although careful attention was observed, as the propeller lifted out of water the engines raced heavily before being throttled by the main valve. As a result, in one heavy race the engine carried away three main bearing caps, the fracture extending right through the center of the oil boxes, as indicated in the sketch.

Under the circumstances there was no time for confusion or any attempt to take various opinions as to the best course to adopt. Repairs were effected immediately in the following manner:

The first step was to knock off the oil boxes flush with the top with a 28-pound hammer. A heavy plank was then fitted to each cap, reaching up to the cylinder bottoms. Over the top end of the plank was tacked a piece of asbestos cloth, in order to protect that end from the heat of the cylinder as much as possible, and in order to press the bearings down firmly wedges were made and driven in under the lower end of the plank and on the top surface of the oil box, as shown in Fig. *B*. These wedges were made of some old lignum vitæ strips which had been preserved from the last time that the stern bush of the vessel had been lined up. These strips were made up into long, thin wedges, which were practically as hard as steel itself. By this means the planks were wedged up firmly and solidly, and enabled the ship to get away, to the great relief of everybody on board. She had lain in the trough of the sea for forty minutes, and lost everything moveable from her decks.

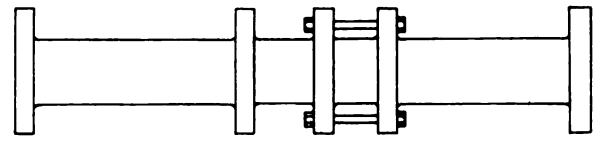
When the engines were got under way again a horizontal plank was fitted between the projecting portions of the oil box which was left and the bunker side, as shown in Fig. *A*, in order to relieve as much as possible the twisting strain on the shaft. The engine was run for a few hours at reduced speed, but after careful watching it was found that the bearings stood all the work which could be put on them, so the engine was pushed up to full speed, and the boat arrived safely at Havre. The surveyor there pronounced the engine perfectly safe to proceed to the Bristol Channel, which was excellent testimony to the efficiency of this hastily rigged-up repair.

Other caps were subsequently obtained from the builders of the engine, who said that in their experience of forty years of commercial life they had never known such an accident to happen before, and, so far as the writer is aware, no similar breakdown has occurred before. It is therefore interesting as showing the very high factor of safety which must be used in the design of such bearing caps, on account of the sudden severe stresses which may be thrown upon them, producing the effect of a heavy blow. At the time of the accident, however, the writer's interest was not centered in the theoretical considerations involved, as his life was never in greater danger than on that occasion.

ANSON C. STOKES.

Repairing a Broken Thrust Shaft.

In repairing a thrust shaft which broke in between two collars of the thrust bearing, two of the thrust shoes were first removed, and four holes were drilled through the collars by means of a hand ratchet. Through these holes four ¾-inch bolts were fitted, and the shaft was drawn together at the fracture. Fortunately, the fracture was diagonal, so that to



LOCATION OF BOLTS IN BROKEN THRUST SHAFT.

some extent the two halves keyed into one another, assisting the bolts in transmitting the stress. For this reason the ¾-inch bolts, which were the largest size which could be found on board, although very slight indeed for this purpose, were not over stressed, and the repair held out satisfactorily for the homeward run of about 100 miles at reduced speed.

FRED CURTISS.

Broken Air-Pump Links.

This fracture is seldom met with in practice by the average marine engineer, consequently few would be apt to know the most logical way to effect repairs that are thoroughly serviceable in every respect. A few years ago such an accident happened on board one of the United States navy colliers, which was originally an English tramp steamer of about 7,000 tons displacement and equipped with the English type of engines. The air, feed and bilge pumps were all connected to the air-pump cross-head, and commonly actuated by the low-pressure engine.

The ship came to anchor for a few hours at Punta Arenas,

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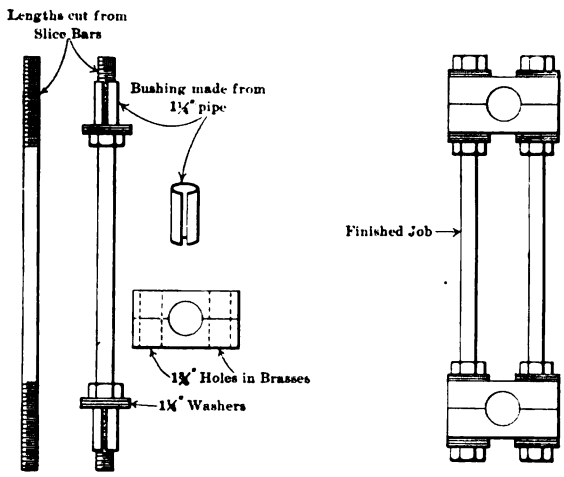
Strait of Magellan, en route to San Diego, Cal., consequently the entire engine crew were busy packing leaky rods, placing liners under eccentric rods, examining the air-pump valves, etc. It was found necessary to insert a few tin liners in back of the tail-rod guide brasses in order to line it out, as the rod had worn the brasses to such an extent that it had considerable play. As the tail rod was screwed onto the threads of the air-pump rod, which projected through the cross-head, it was essential to block up the bucket before the tail rod was unscrewed. After filing and fitting the brasses and lining out the rod, resulting in a good job, orders were given to clear all "gear," etc., away and have everything ready to start.

In the haste which ensued a block of wood was left in the air pump, and as soon as the engine made its first "turn-over" revolution the links bent over and all four snapped off. This, of course, necessitated a further delay of about four hours; nevertheless, very substantial repairs had to be made. Having



AIR-PUMP LINKS MADE FROM SLICE BARS.

no spare steel bars available for the purpose slice bars had to be cut instead. The broken links were disconnected and measured, and four lengths were cut from the slice bars 3 inches longer to allow for washers, etc., as will be explained later. As the diameter of each slice bar was only 1¼ inches, and that of the broken links 1¾ inches, with threaded ends of the same diameter, it was necessary to cut eight pieces of 1¼-inch pipe, splitting them on one side to fit over the threaded ends and take up the space between the 1¾-inch holes and new 1¼-inch rods in all the brasses, as shown in the sketch. After these were roughly filed, allowing no slack when on the rod and in the brasses, threads were cut on all ends to a distance of about 9 inches.

To take the place of the collars on all the rods, a 1¼-inch nut was run down on the thread and three 1¼-inch iron washers were placed on top, then the brasses were shoved on, and secured together by three more washers and another 1¼-inch nut on each end, as will be seen in the sketch.

This repair job proved very serviceable with the engine running at full speed, and was still in good order at the end of two months, when new rods were made in the shop.

J. W. F.

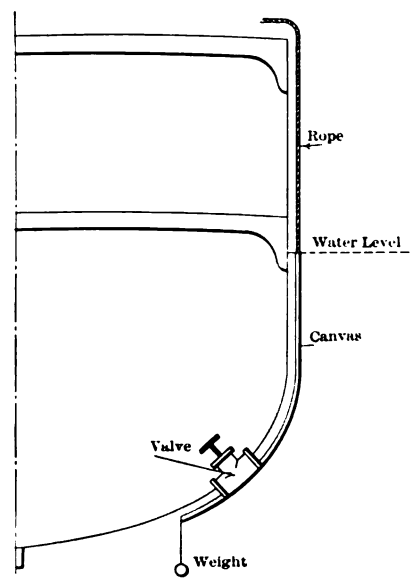
Method of Closing a Large Sea Valve in an Emergency.

We arrived in an English port one night and went below the coal tips to fill up our coal bunkers for a long voyage. The next morning the chief engineer told one of the apprentices to see that the main injection valve on the ship's side (for the port engine) was properly shut, and then to slacken off all the nuts on the main condenser doors and take off the doors preliminary to testing the condenser for any leaky tubes. The first engineer thought he had fully explained the job, and that the apprentice clearly understood what was to be done, so he went into the boiler room to look after the main boilers, which had been opened up for cleaning, leaving the apprentice to carry out the work on the condenser.

In a few minutes we heard a great rush of water in the engine room, and immediately discovered that the apprentice had taken off all the nuts on the main injection valve cover, and that the valve, spindle and cover had been lifted out of place, allowing the sea water to flow into the ship through the full 12-inch opening.

We promptly started all the pumps and called the first engineer, but none of our attempts to plug up the hole was successful. By this time we were up to our knees in water in the stoke-hole, where the donkey boiler was located, and in a few minutes the water would be up to the furnaces. It was necessary to act and to act quickly.

The first officer secured a large piece of awning canvas, and attached a weight to each corner of one end, just heavy enough to sink it. Ropes were attached to the two opposite corners and fastened on deck. The weighted end of the canvas was then dropped over the ship's side in way of the main injection valve. The inrush of water through the valve hole quickly



A BROKEN SEA VALVE EFFECTIVELY CLOSED BY A PIECE OF WEIGHTED CANVAS.

drew the canvas over the hole and completely stopped the inrush of water to the engine room. We then straightened the valve spindle, which had been forced up against an overhead iron beam on the ship's side, and replaced the valve, putting everything back in proper shape. We considered that we were very lucky in that no damage was done by the water either in the stoke-hole or in the engine room. If the ship had been deep in the water with a full cargo it would not have been so easy to fit on the outside canvas.

It seems to me that the first officer deserved great credit for the quick manner in which he decided to try and help his shipmates out of a bad position, and his ingenuity only shows what can be done on board ship when mates and engineers pull together for their employer's benefit.

Some time ago a steamer was sunk in Hong Kong harbor from very nearly the same cause, a mistake having been made by a junior engineer.

Donald McColl.

Shanghai, China.

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Machining a Stern Tube.

Stern tubes are usually made of cast iron, the construction being as shown in Fig. 1. Such a tube is machined as indicated by the finish marks on the sketch. The first step is to fit bridge center pieces at each end, as shown in Fig. 4. Previous to this, however, the casting should be checked over for size. In placing the tube in the lathe, the flanged end should be placed next the chuck or face plate. The chuck is the handiest for this operation, as it affords a solid drive and supports the weight of the tube during the boring-out operations.

The first operation is to rough-turn the end which goes through the sternpost, and then the small bearing surface next

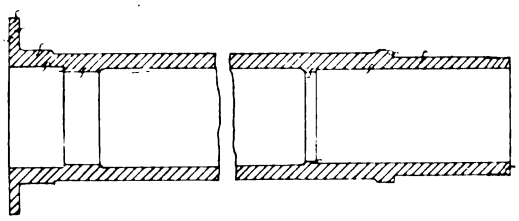


FIG. 1.

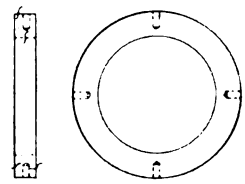


FIG. 2.

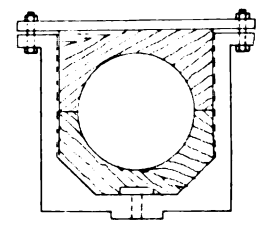


FIG. 3.

the flange at the stuffing-box end; this part fits into the bulkhead. As it is easier to turn the tube to a given size than to bore out the sternpost and bulkhead to gage sizes, these holes are sometimes bored out as closely as possible to drawing size, then gages are made to which the tube is turned.

The distances between the sternpost and the bulkhead are also measured, and the tube machined to suit. This length measurement need not necessarily be very exact, as the flange is not bolted directly to the bulkhead, packing pieces of hardwood being placed between the flange face and the bulkhead. The thickness of these pieces may vary from ½ to 1½ or 2 inches.

The exact length and diameter being obtained, the tube is finished to size and length. At this setting, the inside face of

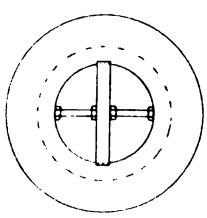


FIG. 4.

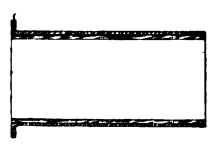
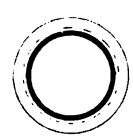


FIG. 5.



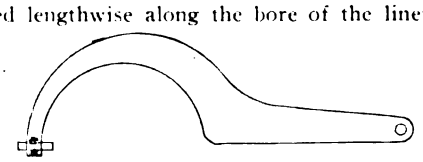


FIG. 6.

the flange should also be machined. As the tube is left one or two inches long to allow for any variation from drawing sizes between the sternpost and the bulkhead, there is often some metal to be taken off at the sternpost end. After the shoulders have been finished to length, this part should be roughed down to clear the nut for the end of the tube; threading the tube to fit this nut being the next operation.

This nut is shown in Fig. 2. It is simply a round iron ring threaded internally, four holes being bored, as shown, for tightening and unscrewing. The spanner for this nut is shown in Fig. 6; the pin which fits into the holes being simply a piece of mild steel of suitable diameter, which need not be fastened into the spanner in any way.

After fitting on the nut, the next step is to bore out the tube for the brass liner shown in Fig. 5. Still keeping the tube on the lathe centers, the steady rest (Fig. 3) should be fitted. This is simply two hardwood blocks—beechwood being very suitable—fitted into a cast iron support. After fitting the blocks to the tube and screwing up the bolts, the lathe center should be slacked back a short distance and the lathe run for a short time to see if the tube sinks down into the rest. Grease or tallow should be used to lubricate the wood blocks. The surplus metal should now be cut off, and the tube finished to length.

The inside is now bored out to gage size, a suitable boring bar carrying a double-ended cutter being used for the smaller sizes, while for the larger ones a suitable head carrying four tools may be used. This completes the turning operations on this end.

The tube is now reversed in the lathe. As the bearing strip at the flanged end is larger in diameter than the part which fits into the sternpost, new bearing blocks must be fitted, the tube being supported on the bridge center. After fitting the rest, and still keeping the bridge center, the outside edge may be turned and the flange faced up to the proper thickness. The bridge piece is now removed and the tube bored out to gage size for the stuffing box and neck ring. This completes the turning operations on the tube.

The neck ring and gland are now machined to size, but as they are simple turning operations, no description is necessary. The neck ring is made of brass; the gland of cast iron and bushed with brass.

In turning the liner which fits into the outer end of the tube, the outside diameter is made a nice driving fit for the tube, the bore is machined to drawing size, and need not be very exact, as lignum vitæ strips are fitted into it for the shaft-bearing, this wood being very suitable for journals under water. To facilitate fitting in the wood, a small brass strip is pinned lengthwise along the bore of the liner, being less in thickness than the finished size of the wood. As it is kept toward the top of the tube, it forms a channel for lubricating the bearing.

The liner is again placed in the lathe and the wood bored out to fit the liners on the tail shaft. It should not be made too nice a fit, because if the tube should be lying outside in the wet some time before fitting the shaft and tube into place in the ship, the wood may swell enough to cause considerable trouble. Before being driven in, the liner is coated with thin red lead. Holes are drilled and tapped into the sterntube, through the flange of the liner, and pins fitted in to keep the liner in place. Drilling, tapping and fitting in the studs for the gland, and also the pop on the tube (not shown) for the lubricating arrangement, complete the machine work. W. Burns.

A Unique Experience.

The steamers *Grand* and *Rapids* had a unique experience about the first of May, when an attempt was made to take them through the locks at Kaukauna, Wis. After the gates

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were opened the boats were drawn partly into the lock by suction, but there it was found that the breadth of the beam and width of the lock were approximately the same, each being about 35 feet. The ships stuck to the side of the locks and would not move. After investigating, Captain J. F. Cavanaugh decided to grease the sides of the ships. The grease was applied and the boats passed through.

Trouble with the Main Steam Pipe.

The steam-pipe system on board a boat is perhaps one of the most important features of its equipment, and yet one which is liable to the least amount of inspection and, it would appear, the greatest defects in design. It is an extremely difficult thing, owing to the small amount of space available for the purpose, to adequately arrange the piping so that it shall be properly trapped at every point, and so that adequate precautions are taken to ensure absence of undue strain due to expansion and contraction. Moreover, in spite of regulations as to safety, it sometimes occurs that either by accident or for cheapness material is put in which is not quite up to the standard, and even, if in the first place, the workmanship and material are quite good, the boat is sometimes run past the limit of safety in point of time without adequate overhauling and inspection.

These remarks may apply to some extent to an accident which occurred to the main steam pipe on board a boat known to the writer. This pipe was made of copper and had no expansion joint. There were, however, two bends in it, in order to allow a section of the pipe to run at a lower level, and these two bends were considered sufficient to allow for expansion. On one voyage, however, the steam pipe opened on its longitudinal seam at one bend, and a little further along the length the pipe had cracked opposite to the seam at the other bend. The trouble was first detected by the discovery of a hissing sound from the top of the boilers, and as the pressure was at the time right up to its full working point, or at 160 pounds per square inch, an examination was made from a safe distance. Steam was seen issuing from the defects in such a large volume that it was easily seen that the size of the fracture was considerable. For this reason the fires were at once checked, and the engines opened out to their fullest possible extent. At the same time extra water was pumped into the boilers and the feed heaters were shut off. These operations were all done as nearly as possible at the same time immediately after the discovery of the trouble, all the engineers hastening to minimize the trouble. As soon as the pressure was lowered sufficiently, that is to say, to about 125 pounds, the main stop valves were closed, thus cutting off the escape of steam from the fracture. As the way to the main stop valves lay over the defective pipe the reason for lowering the pressure is obvious.

After steam was shut off a careful examination at close quarters was made and two cracks were discovered, one about 3 inches long and the other about 5 inches. In order to repair these a Muntz metal patch was cut out and bent to fit the pipe at each of the fractures. This patch was made of 1 1/16-inch stuff, about 3 inches wide, and long enough to lap over the ends of the cracks. Clamps were then made of strap iron, ¼ inch by 1¼ inches, and bent so as to clamp round the pipe. By means of these the patches were very strongly secured to the faulty pipe. Three clamps were used for the smaller crack and four for the larger one. Each patch was put on the pipe over a joint made of rubber jointing, and the whole collection was then tightened up by means of the clamps. After this iron-seizing wire was wound over the entire length of both the patches, and also for a good distance beyond the ends, in order to strengthen the pipe and prevent the cracks from extending. Steam was then admitted carefully to the repaired pipe, and it was found that a satisfactory repair had been executed, no further trouble being encountered. This may be considered as a fairly quick repair job, as the ship was stopped only for a matter of two hours, and on restarting everything was found quite right.

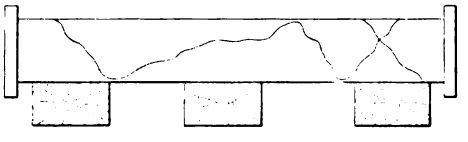
After a run of about twelve days, on arrival of the boat at port, the pipe was condemned and a new one fitted. After removal from position the condemned pipe was found to have deteriorated considerably, as some pitting was discovered on the interior. Moreover, the metal was very brittle. This defect is a somewhat common cause of trouble in copper pipe which is exposed to high temperatures and alternate compression and expansion, due to changes of temperature, which should make engineers somewhat careful as to its use without periodical inspection.

Chas. P. Starkweather.

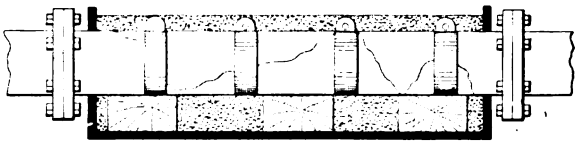
Trouble with Frozen Pipes.

One source of worry which the engineer of a ship trading in cold climates will find is the disastrous effect of frost in bursting pipes containing water. No matter how careful he may be in seeing that all pipes and vessels are systematically drained, he is liable to be caught now and again, and a considerable amount of trouble may be caused. One of the most frequent locations of trouble of this nature is in the tank-filling pipes along the tunnel. These, if not properly attended to and drained, are almost sure to contain water. Moreover, if the ship is by the head or stern, and the aftermost tank valve is shut down, water will be lodged in the pipe.

The best plan in order to make sure of things in frosty weather is to break a joint in the pipe. As a rule, there is a section of pipe in such a system about 4 or 5 feet long, which



PIPE ON BLOCKS



PIPE WITH CLAMPS IN CEMENT

is made of cast iron, and this is usually the member that is found in from ten to twenty pieces when a thaw comes. As it often occurs that the ship is making for a port where a new pipe would cost a good deal of money, and as, moreover, the captain may be in a hurry to run up the tanks, it is useful to bear in mind the following hint for repairing such a pipe, no matter how many pieces it may have been broken into:

These pieces should be brought into the engine room and carefully placed on suitable chocks of wood, fitting in the pieces until the pipe is complete. When this is done clips or bands should be made, in order to hold the pipe together, so that it can be lifted in one piece. This requires in some cases considerable ingenuity, and, of course, it will not be air-tight. When, however, it is in this position it can be taken back along the tunnel and rejointed in the usual way. After this a rough box or trough is made, taking in the full length of the broken pipe and covering all the fractures. This must then be filled in with Portland cement, and it will be found that after this has set the pipe will be perfectly tight and capable of doing the work that is required of it.

J. M. Speedwell.

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Repairing the Crank Shaft of a Triple Expansion Engine.

The mishap occurred in a large tramp steamer, on a voyage from Argentine to the Continent. The intermediate-pressure crank twisted round on the after portion of the shaft, finally leaving it and stopping near the bottom center, at about 120 degrees from its proper position.

The valve casings were arranged at the forward ends of each cylinder, and the shaft consisted of three cranks, all exactly alike in every respect; a very fortunate arrangement, as it turned out. The three were fixed at the usual angle, 120 degrees apart, each crank shaft carrying its own eccentric.

Each crank shaft consisted of five pieces: crank pin, two webs and two short pieces of shafting, with coupling ends. The different pieces were shrunk together in the usual way, and were further secured by 1¼-inch diameter dowel pins, fitted half in the webs and half in the shaft. In passing, the writer would suggest that these pins should be frequently examined, as he has met with several cases where they have worked slack.

The after web of the intermediate-pressure crank shaft had been imperfectly fitted, and in service the dowel pin had loosened, until it finally dropped out, being subsequently found in the crank pit. At the time of the accident the engines were running freely, and there was no reason to suspect trouble. Then, without any warning, the engine made a few spasmodic jerks, and then stopped, with the cranks in the position mentioned above, the low-pressure crank in its usual position and the intermediate-pressure and high-pressure 120 degrees away. Steam was roaring through the low-pressure relief valves.

It was some little time before the relative position of the cranks was noticed, as on the first preliminary examination everything appeared to be in its proper place. Then the best part of a day was taken up in attempts to move the low-pressure crank round to the others, by the aid of turning gear, screw-jacks, etc., but without success. If this could have been done it was intended to drill several holes and fit dowel pins in them to hold the cranks together.

Finally it was decided to remove the defective intermediate-pressure shaft, and replace it with the high-pressure shaft, which, it will be remembered, was the same size in every respect. This was done, fresh keyways being cut and the eccentrics fitted on the new shaft to meet the new conditions of the valve gear. The high-pressure valve was taken out, and steam allowed to flow through the exhaust port to the intermediate-pressure casing. The high-pressure valve spindle was lashed and left in the gland, and the gland tightened to prevent leakage. As the high-pressure connecting rod, valve gear, eccentrics, etc., were removed when the engine was stripped to change shafts, the high-pressure piston was lowered to the bottom of the cylinder, and the piston gland tightened up.

The trouble occurred at 7 A. M. one morning, and the repairs were completed by 5 P. M. the next day, or after thirty-four hours' continuous work. The arrangement answered very well, and the job was a very creditable one. The engineers were assisted the whole time by the deck hands, who tackled and handled the gear as it was disconnected.

There was a spare length of crank between decks, but it could not be got at, as there was a grain cargo. If it could have been used there would have been a good deal of trouble, and it would have taken up a lot of time rigging gear to strike it down into the engine room.

On restarting, the boiler pressure was reduced from the original 180 pounds to 100 pounds, but it was found more economical and to give easier steaming to regulate the throttle valve, so as to obtain about 65 pounds in the intermediate-pressure valve casing and 15 pounds in the low-pressure. The revolutions were reduced by 6 and the coal by 2 tons.

On arrival at the Continental port it was decided to leave the engine as it was till the vessel's arrival in England, where the faulty shaft was replaced.

Repairs to Sea Cocks, Etc.

Engineers on board ship have frequently to execute repairs by means of expedients which would not be thought of on shore, and they are handicapped by the fact that they do not always have a proper equipment of spare parts and workshop tools. One of the most common troubles is the leakage to be found in steam or water cocks, the plugs of which are very apt to corrode. This makes it impossible to keep them continually tight, and if frequent grinding is resorted to a considerable amount of metal would be worn away from the box, depreciating its life considerably.

A very good plan of executing a repair of this nature, if the ship is running in a trade where shore repairs are expensive, is as follows (it applies to sea cocks of all descriptions, no matter what the size of the plug may be): After withdrawing the plug from the cock, it should first be cleaned free of all grease and foreign matter, and then tinned all over with solder in the usual way. The next step is to cut out a paper

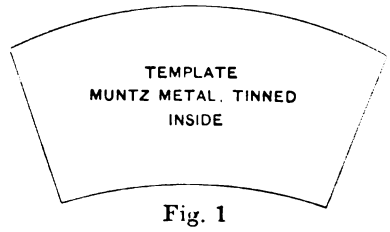

Fig. 1

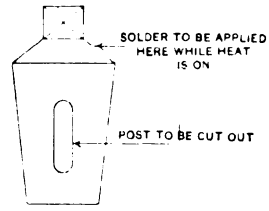

Fig. 2

template to go round the plug, and making a butt-joint on a portion of the cock away from the ports; the shape of this template would be as shown in Fig. 1. A sheet of Muntz metal, 1/32 inch thick, is then cut to suit the template, and then bent round the plug and fixed to it by means of copper wire so as to fit accurately, and also to close together on the butting surface. When this is done the whole plug is brought to a heat sufficient to melt the solder, and at the same time sufficient solder is run round the neck of the plug to sweat the sleeve. When the plug is cooled again, it only remains to cut out the ports from the sleeve and file off any humps that may be found, and then to grind the cock in its place in the box. In this way the abrasion of metal which occurs from repeated grinding, and which would eventually make the cock too small for the box, is virtually added on to the cock, giving the gear a fresh start. In order to bring the plug to a good heat sufficient to melt the solder, either a paraffin blow lamp may be used or else a large nut heated to a blood-red temperature may be placed over the plug. Alternatively, the fire-bar tongs heated up will serve the same purpose. George Halsey.

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Fractures in Propeller Shafts.

A very serious accident which might easily cripple a boat while at sea is the fracture of the shaft which passes through the tunnel to the propeller. It very much depends upon the nature of the fracture as to the course which should appropriately be adopted, and the repair would be very much complicated if the shaft were bent. If, however, it is a clean fracture without bending it can be repaired as follows:

The ends of the fracture should be brought firmly together, and then a hole about 1½ inches in diameter should be drilled clean through the center of the break. A pin should then be made to fit into the hole, and this really forms a key between one part of the fractured shaft and the other when it is set turning again. The ends of this pin should be kept flush with the surface of the shaft over the break. Some bars of iron, 2½ or 3 inches thick, should then be obtained and cut to length of about 12 or 14 inches. It is advisable to cut keyways parallel with the axis of the shaft, passing from one-half of the break to the other in order to hold these bars, as

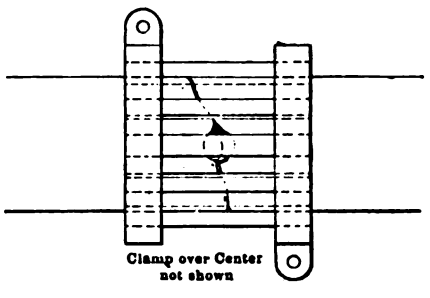

FIG. 1

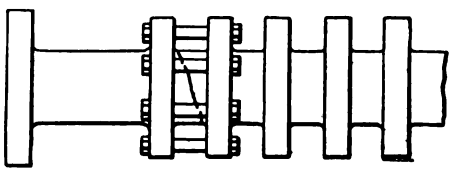

FIG. 2

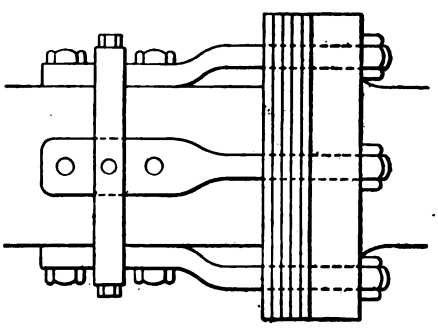

FIG. 3

VARIOUS METHODS OF TEMPORARILY REPAIRING A FRACTURED PROPELLER SHAFT.

shown in Fig. 1. The keyways should be about ½ inch deep, and one of the bars should lie over each end of the pin, so as to keep it in position. Over each end of the series of bars thus placed round the shaft strong iron clamps should be bolted, as shown, or; if this is not possible, the bars should be pinned onto the shaft by drilling tapping holes into the shaft, and putting screwed pins through the bars and into the solid metal of the shaft. It is preferable, however, to have the clamps, which can be made by means of the portable forge.

With the aid of this repair it will be found that the boat can be allowed to proceed, although care should be taken in starting the engines to avoid sudden strain on the shaft. It should be carefully watched should the boat be running in a following sea, as any lifting of the propeller out of the water and sudden dipping in again will impose severe twisting strains on the shaft, which will soon loosen the repair.

Should a fracture occur in the section of the shaft between the thrust collars the trouble is rather difficult to get at, but may be repaired by drilling tapping holes through two of the collars adjacent to the fracture. The number and sizes of holes should be sufficient to afford enough metal to stand the twisting strains imposed on the shaft. The bolts should be driven into place and screwed up tight, so as to join the edges of the fracture firmly together. The engine will now be able to drive the shaft with the remaining collars, but inasmuch as there is now less bearing surface interposed to take the endwise thrust of the propeller it is obvious that the speed of the engine will have to be reduced in proportion to the number of thrust collars which have been thrown out of action by the repair. This breakdown is illustrated in Fig. 2.

Sometimes the fracture occurs at one of the flanges which couple the various sections of the tunnel shaft together, and one of the forms of fracture is the complete shear-off of the flange from the shaft. This at first sight appears to be an accident which requires more than an ordinary seagoing equipment to overcome, but Fig. 3 gives an idea of how, in practice, the repair may be made sufficiently strong to enable the ship to be brought home to port. The end of the broken shaft should be squared up so that it will abut accurately against the unbroken flange end to which it is adjacent. Then the squared-end plates should be fitted to make up the thickness of the broken flange, so as to give the distance along the length of the tunnel. The coupling bolts belonging to the flange coupling are now taken, and lengths welded onto them of sufficient extension to allow tap bolts to be fitted so as to bolt onto the fractured part of the shaft. This extension should then be clamped onto the shaft and tap bolts placed over the clamps. It will be seen, therefore, that projecting out from the broken part of the shaft and through the distance plates will be the requisite number of screwed rods which will go to form the coupling bolts. The two sections of shaft can then be coupled up in the usual way, and if care is taken to start up the engine easily it will be found that the repair will carry the boat through.—J. A. Seager, in *Power and the Engineer*.

Newcastle-on-Tyne, England.

Decrease in American Shipbuilding.

Reports from the Bureau of Navigation show that during the year ended June 30, 1909, 1,362 merchant vessels of 232,816 gross tons were built in the United States and officially numbered by the Bureau of Navigation, compared with 1,506 of 588,627 gross tons during the fiscal year 1908, which was the record year of American shipbuilding. This year's output was the smallest since 1898; but shipbuilding contracts indicate a material increase during the new fiscal year.

On the Great Lakes thirty-six steamers of 88,436 gross tons were built, including the *Shenango*, 8,047 tons, the largest vessel ever built on the Lakes. Only two ocean steamships, *Mars*, 5,451 tons, and *Mohawk*, 4,683 tons, were built. The *Edward B. Winslow*, Bath, Me., 3,424 tons, is the largest wooden schooner ever built. No vessels for foreign trade and no square-rigged vessels were built during the year. Of the year's output, 60,952 tons were barges and canal boats.

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Steam Engine Indicators for Marine Work.

With this issue we begin publication of a series of short, illustrated articles describing various types of steam-engine indicators suitable for marine work which are manufactured in Great Britain and America. These articles are published as the result of many inquiries from our readers who have become interested in Lieutenant Root's valuable series of articles on the indicator, which was begun in the July issue. It is not our intention in the brief descriptions of commercial indicators to present the manufacturers' claims for these instruments, but simply to publish the facts regarding the design and construction, pointing out the reasons for various modifications and their probable effect on the accuracy of the instrument. A careful perusal of these, and a clear understanding of Lieutenant Root's comprehensive article, should give the average marine engineer a thorough knowledge and understanding of the requirements of a good indicator for marine work, so that he can judge for himself what type is necessary to suit his own particular needs.

Notes on Producer-Gas Operation.

Small marine producer-gas plants are more extensively used in Holland at the present time than in any other country. They are used chiefly on barges and canal boats, where it is impossible to employ experienced engineers to care for the plants; and since these are somewhat the same conditions under which it is expected that the immediate development of producer-gas for marine work will take place in other countries, it is interesting to note the successes and failures which the bargemen of Holland have had with these plants. Elsewhere in this issue we publish a thoroughly practical article describing the construction and operation of these plants. The author points out that, contrary to the general impression, a marine suction gas plant is a very sensitive power plant to operate, as compared with one employing steam or oil engines. The ordinary troubles which are encountered in the operation of any gas engine are likely to be augmented when producer-gas is used as the fuel by additional troubles with the producer itself. It is, therefore, absolutely necessary that the man in charge should have a good, practical knowledge of the working of the plant in order to operate it successfully, although he need not be an experienced engineer or mechanic.

Without going into the details of the particular plant illustrated and described, the following points should be noted with regard to the operation of a producer-gas plant: The producer itself is likely to fail because of insufficient gas when starting, a bad quality of gas, or no gas while working. The first will happen if the motor is started before the producer is completely charged and combustion well under way. The second the author attributes to a lack of hydrogen in the gas. However true this may be as regards the type of plant used in Holland, it should be remembered that there are many opinions regarding the value of hydrogen in producer-gas. Many engineers consider not only that hydrogen is not an essential constituent of producer-gas, but that its presence in the explosive mixture involves a distinct loss, due to the formation of water vapor when combustion takes place. The third difficulty is usually caused by clogging of the gas passages in some part of the apparatus, and should be avoided by careful operation.

As far as troubles with the motor are concened, the reader will do well to note carefully the points brought out by Mr. Smith in his paper on "Gas Engine Construction for Producer-Gas Use," an abstract of which is published on page 387. Doubtless there are some

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things in this paper which will be severely criticised; but, on the whole, Mr. Smith brings out very strongly the main requisites for successful operation on producer-gas, and it is likely that troubles will be due to the fact that these points have either been ignored or not fully understood. Whatever opinions are held regarding the value of high compression and the effect of a large percentage of hydrogen in the explosive mixture, it is certain that an ample and effective ignition gear should be provided; that the piping and valve passages should be of sufficient size and without abrupt bends, and that the compression space in the engine should be as small and free from openings or pockets as possible.

Combined Reciprocating and Turbine Machinery.

The combination of reciprocating and turbine machinery was suggested by Mr. Parsons primarily to increase the power obtainable by the expansion of steam beyond the limits possible with reciprocating engines. There was no doubt that this object could be attained, and that increased power for a given steam consumption would result; but it remained to be proved that such a system of combined machinery could be efficiently employed for the propulsion of a ship, since it involved the use of at least three propellers and the determination of such questions as what percentage of the total power should be applied to the turbine-driven propeller for the most efficient results, etc. Of course, such a system of combined machinery was designed for use only on vessels of moderate speeds, for which turbines alone would not be efficient.

The first merchant vessel to be fitted with this system of propelling machinery was the *Otaki,* of 9,900 tons deadweight carrying capacity and 12 knots designed speed, built by Messrs. Denny, of Dumbarton, for the New Zealand Shipping Company. The *Otaki* is virtually a sister-ship of the twin-screw vessels *Orari* and *Opawa,* previously built by the Messrs. Denny for the same company, and fitted with reciprocating engines. These vessels have given a splendid chance for comparison of the two systems of propulsion, and the results obtained, both on trial and in regular service, have recently been made public by Engineer-Commander Wisnom in a paper read at a joint meeting of the Institution of Engineers and Shipbuilders in Scotland and the North East Coast Institution of Engineers and Shipbuilders.

On her trials the *Orari* obtained a mean speed of 14.6 knots on the measured mile at Skelmorlie, while the *Otaki,* under the same conditions, with apparently greater ease obtained a mean speed of over 15 knots on a total water consumption per hour of 6 percent less than the *Orari,* while the total water consumption per hour in the *Otaki* at 14.6 knots was 17 percent less than in the *Orari* at the same speed. The proportion of total power developed in the turbine of the *Otaki* was found to vary with the speed. At full power this proportion was about one-third, while at very low speeds the turbine was doing only a small proportion of the work. At a speed of 14.6 knots the indicated horsepower in the *Orari* was 5,350, and the corresponding power in the *Otaki* was 5,880. At this speed the effective horsepower was 3,210 in the *Orari* and 3,350 in the *Otaki,* the propulsive coefficients being thus 60 percent and 57 percent, respectively. The propulsive coefficient in the *Otaki* at full speed fell to 54 percent.

On account of the distinction between the shaft horsepower and indicated horsepower in two vessels driven respectively by reciprocating engines and steam turbines, it is important not to rely wholly on the foregoing figures, which simply give a percentage based on the total water consumption and speed. Comparing the water consumption per effective horsepower per hour, the *Otaki* showed a gain of 20 percent, and again comparing the water consumption of the two vessels per indicated horsepower, taking the indicated horsepower for the *Otaki* as that obtained in the *Orari* for corresponding speeds, the gain in the *Otaki* was 17 percent.

In addition to the trial results, Mr. Wisnom gave some figures covering the performance of the vessels in actual service. On the voyage from Liverpool to Teneriffe the coal consumption of the *Otaki* was 11 percent less than the means for the sister-vessels, *Orari* and *Opawa,* under similar conditions, and at practically the same speed. A careful comparison of the coal consumption of the *Otaki* for the round voyage with that of the sister-ships on similar voyages at the same speed shows an apparent gain of about 8 percent. This figure, however, it is expected will be bettered.

The dimensions of the engine room in the *Otaki* were the same as in her sister-ships, and were found sufficient to admit of an arrangement of the machinery which gives satisfactory access to all working parts. The total weight of machinery in the *Otaki* was about 30 tons more than in her sister-ships, or an increase of about 3.25 percent. In view of the greater economy of the combination system, the boiler power of the *Otaki* might have been reduced. If this had been done, the total weight of the combination system would not have exceeded that for reciprocating engines, while the saving in bunker capacity would more than have balanced the loss in cargo-carrying capacity due to the three shaft tunnels. These losses in the *Otaki* were compensated by increasing the length slightly.

On the whole, the performance of the *Otaki* seems to bear out the expectations that a high degree of economy may be expected with this form of machinery; that is, that in vessels of low speeds additional power can be obtained for the same water consumption; but the problem still remains in each case of how to utilize this power most efficiently; that is, how to make the three propellers as efficient as twin screws, and what proportion of the power to develop in the turbine.

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Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Tons.	Knots.		Aug. 1.	Sept. 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons.....	96.5	98.0
Michigan ...	16,000	18½	New York Shipbuilding Co...	99.4	100.0
Delaware ...	20,000	21	Newp't News Shipbuilding Co.	91.8	94.8
North Dakota	20,000	21	Fore River Shipbuilding Co..	90.3	93.5
Florida	20,000	20¾	Navy Yord, New York.....	24.8	29.2
Utah	20,000	20¾	New York Shipbuilding Co...	33.2	38.1
TORPEDO-BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons.....	95.6	**96.4**
Lamson	700	28	Wm. Cramp & Sons.....	88.4	90.3
Preston	700	28	New York Shipbuilding Co...	90.1	93.0
Flusser	700	28	Bath Iron Works.....	90.0	92.4
Reid	700	28	Bath Iron Works.....	84.2	89.6
Paulding ...	742	29½	Bath Iron Works.....	21.7	27.3
Drayton	742	29½	Bath Iron Works.....	20.7	24.6
Roe	742	29½	Newp't News Shipbuilding Co.	57.4	60.9
Terry	742	29½	Newp't News Shipbuilding Co.	51.9	58.5
Perkins	742	29½	Fore River Shipbuilding Co..	44.6	51.7
Sterrett	742	29½	Fore River Shipbuilding Co..	41.2	48.6
McCall	742	29½	New York Shipbuilding Co..	22.5	25.3
Burrows	742	29½	New York Shipbuilding Co..	22.4	25.4
Warrington..	742	29½	Wm. Cramp & Sons.....	32.7	39.1
Mayrant	742	29½	Wm. Cramp & Sons.....	37.2	45.2
No. 32.....	...	....	Newp't News Shipbuilding Co.	0.0	0.6
No. 33.....	...	....	Bath Iron Works.....	1.1	2.2
No. 35.....	...	....	Fore River Shipbuilding Co..	0.8	2.5
SUBMARINE TORPEDO BOATS.					
Stingray	...	..	Fore River Shipbuilding Co..	95.0	97.5
Tarpon	...	..	Fore River Shipbuilding Co..	95.0	98.8
Bonita	...	..	Fore River Shipbuilding Co..	90.6	92.0
Snapper	...	..	Fore River Shipbuilding Co..	87.6	92.0
Narwhal	...	..	Fore River Shipbuilding Co..	94.5	98.2
Grayling ...	...	..	Fore River Shipbuilding Co..	90.6	91.8
Salmon	...	..	Fore River Shipbuilding Co..	81.8	83.7
Seal	...	..	Newp't News Shipbuilding Co.	23.2	24.8
Pickerel			The Moran Co.....	4.3	8.4
Skate			The Moran Co.....	4.3	8.5

ENGINEERING SPECIALTIES.

Clifton Marine Gasoline (Petrol) Engines.

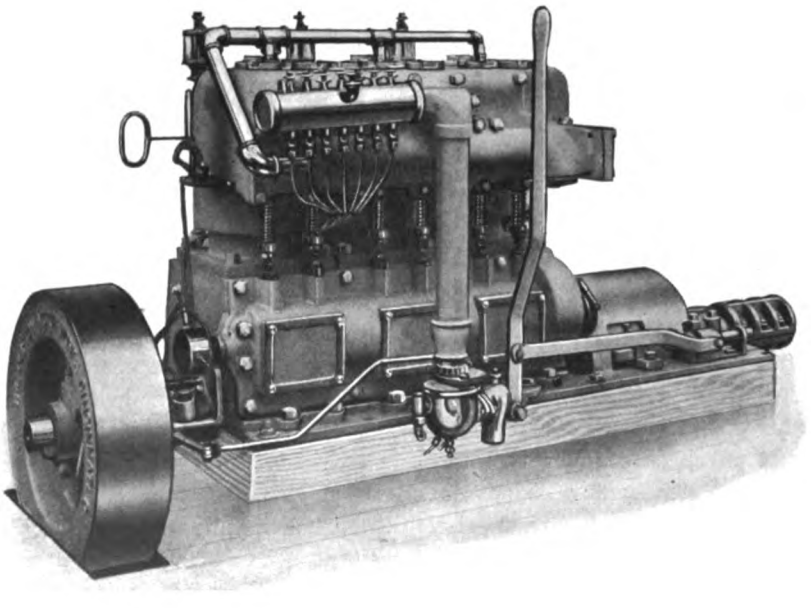
The Clifton marine engine, manufactured by the Clifton Motor Works, Cincinnati, Ohio, has been on the market for a number of years, being used largely in the Southern and Western States along the Gulf of Mexico and its tributary bays and streams. These engines are suitable for heavy sea-going vessels, river boats, lake vessels and cruising launches. The Clifton engine is one of the simplest four-cycle engines on the market, the main idea in the design being to bring the motor down to the fewest number of parts and to make these parts extra strong and substantial. As a result, this company claims never to have had a broken crank shaft or connecting rod, and that the reversing gear, which is also very substantially built, has stood the test of practical service without failure.

The cylinders are cast with hoods integral with the cylinders, so as to do away with packing. The design is such that there is no packing under pressure anywhere in the engine. The exhaust is water-jacketed along the entire length of the engine, and at the end of the exhaust valve a small stream of water is admitted, which cools down the exhaust gases so that there is no danger of the exhaust pipe setting fire to the boat. Simplicity is also obtained in the design of the water pump and circulating system, and all passages are large and unobstructed. The engine is equipped with an automatic throttling governor, to prevent it from racing when the clutch is disconnected. This is a point which will be greatly appreciated in boats which are operated by one or two men, where the pilot frequently operates the clutch on the engine.

A Universal Plate, Bar and Angle Shear.

To cut angles or plates on an ordinary punching and shearing machine requires considerable changing, which consumes time. A machine which is always ready for any kind of work, and is therefore a great labor-saving tool, is placed on the market by the Covington Machine Company, Covington, Va. In this machine a plate shear is placed on the front of the machine with angle shears in two square openings in the side. The latter will cut right and left angles of even or uneven legs to any angle up to 45 degrees. The shears are all driven from one pulley or motor. The angle shears, with their knives, travel at angles of 45 degrees with the horizontal, and the plate and angle shears can be operated singly or together. A patent clutch mechanism, which contains no springs to keep up a continual knocking of the jaws, is arranged to positively stop at the highest point of the stroke. The clutch lever is universal, and can be swung to any position to suit the operator.

It has always been a difficult matter to cut angles with uneven legs, unless large, special machines, too expensive for ordinary manufacturers, have been installed for this purpose. It is claimed that this shear embodies all the best features of a double-angle shear, and also the best features of a plate and bar shear. Bars may be cut by either the angle or plate shear.



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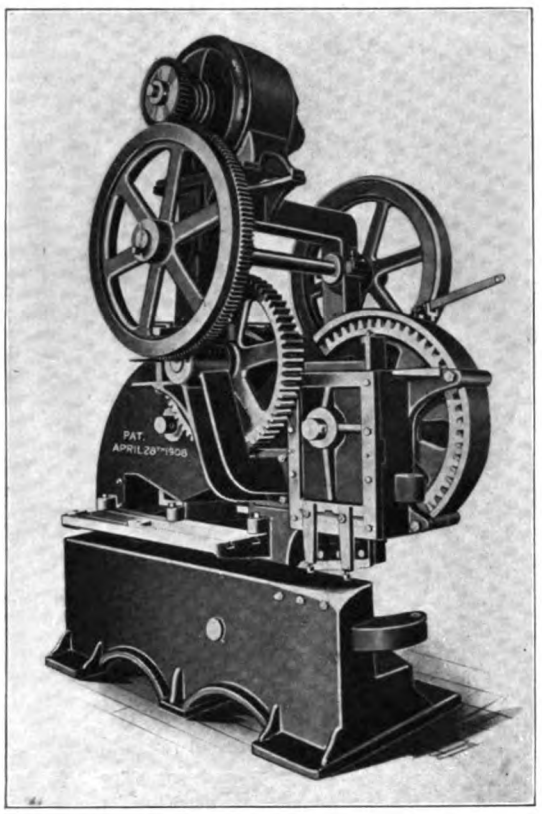
The machines are designed to occupy the minimum of room, are strongly geared, and of massive design throughout. The gear covers, besides acting as safety guards, form the bearings



for the gears, and the gears in turn form the bearings for the eccentric shafts, so that by this method a maximum stiffness of design and protection for the workmen are secured.

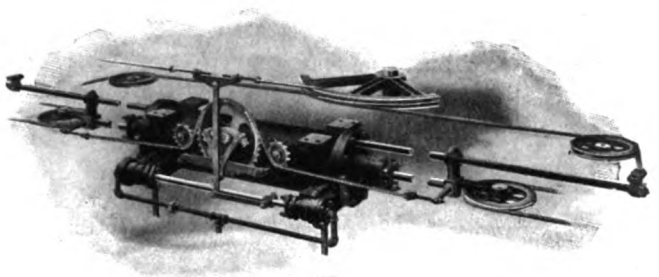
An Improved Steering Engine.

The failure in the past of steering engines in which the movement of the tiller ropes was taken directly from a reciprocating piston instead of from a revolving drum, has been due entirely to their valve mechanism. Either the rudder movement was too sudden, or the rudder could not be compelled to retain the desired position perfectly and indefinitely. The Nash-Century steering engine, which we illustrate, is made with a novel valve motion, by means of which, it is claimed, these objections are completely overcome. The engine is free from a multiplicity of wearing and power-absorbing parts. The piston rod is prolonged equally from both ends of the steam cylinder and terminates in sheaves, through which the tiller ropes pass in a way to give them a travel at the quadrant equal to twice that of the piston. Steam pressure admitted near each end of the cylinder acts upon both ends of the piston to hold it in position, and at all times balances it against the pressure on the rudder.

The valves which admit pressure to the cylinder are of the piston type and have a common spindle. When the controlling gear is stationary, these valves are in mid position; that is, the valve rings cover the steam ports with slight overlap and prevent admission of steam to the cylinder.

The operation of the steering wheel to port or starboard as desired causes the control lever to move. This control lever, swinging about the floating fulcrum, transmits its pull to a connecting rod, the other end of which communicates with the valve spindle. The resulting movement of the valve spindle uncovers the ports, admitting steam through the valve at one end of the cylinder and exhausting through the valve at the other end. The unbalanced cylinder pressure then moves the piston rod, and thus, by means of the sheaves attached to the ends of the piston, the cables to the rudder quadrant are moved, imparting corresponding movement to the rudder.

At the same time the movement of the piston causes the block chain attached to the sheave blocks at each end of the piston to rotate the sprocket and its concentric pinion. The sprocket and pinion are keyed to the same shaft and are practically one piece. The pinion meshes with an arc which swings about a center on a bracket on the cylinder casting. Extending from the center pivot of the arc and diametrically opposite to the toothed part is a projection which forms the fulcrum for the control lever. This explains why we previously spoke of this fulcrum as "floating." The movement of the arc makes the fulcrum travel in a radial direction and opposite to that taken by the valve spindle in opening for steam.



Thus the instant the piston has reached the desired position it has, by its own action as transmitted through piston rods, chain, sprocket, pinion, arc, fulcrum and connecting rod, automatically closed the valve and shut off the steam. To keep the piston moving it is consequently necessary to keep the control lever moving by operating the steering wheel. In other words, the steering wheel operates against an automatic cut-off, which constantly acts to prevent movement other than that desired as indicated by the steering-wheel dial.

In the event of undue strain being put upon the vessel's rudder, it is claimed that the engine instantly operates as a buffer. The action is just the reverse of that necessary for steering, but accomplishes the same purpose, that of keeping the rudder exactly where desired. The movement of the rudder forces the piston to travel. The travel of the piston opens the piston valve at the end of the cylinder towards which the piston is traveling. Steam is thus admitted in proportion to the movement of the valve, and by the time the piston has been driven back to its proper position the steam is again automatically shut off.

This gear is so sensitive that the turn of one spoke of the pilot wheel will perceptibly alter the course of the boat. The action is perfectly noiseless, as the piston and other working parts move slowly. The steam cushioning of the rudder is also claimed to be a great advantage, as it makes the rudder safer from damage in heavy seas, or from sudden impact with solid objects, and does away with the necessity of spring-cushioning the quadrant.

The Nash-Century engine is intended as a "steam only" machine, but, where desired, provision may be made for hand steering also. It is manufactured by the Century Engineering Company, Ogdensburg, N. Y.

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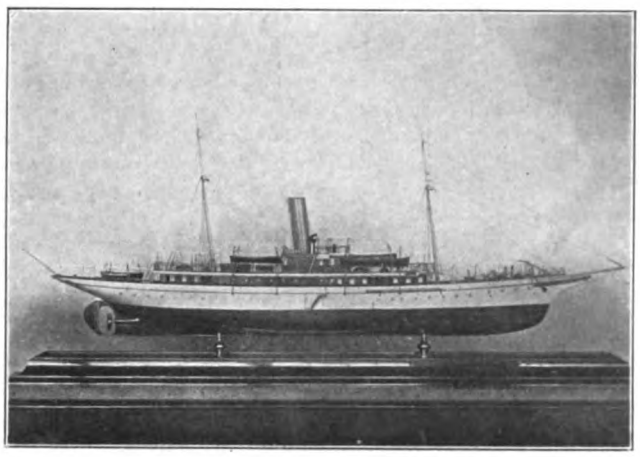
Kelso Models.

The oldest firm of model makers is the Kelso Company, Glasgow. Originally the firm made optical and philosophical instruments, but it soon undertook the building of ships' models and the manufacture of various small articles that go into ships' models, such as engines, fittings, etc. The models are usually made on a scale of a ¼ inch to the foot, although



sometimes to a scale of ⅛ or 1/16 inch to a foot, from drawings supplied by the shipbuilders. Among the models which have been made by this firm are those of the famous old Clyde tea clippers, the *Shamrock* yachts, and at the present time the models of the new giant White Star liners *Olympic* and *Titanic*. From the illustration it will be seen that the models are complete in every detail, including rigging, deck fittings, etc. Recently this company has done much model work in connection with experimental towing tanks.

INDICATORS.

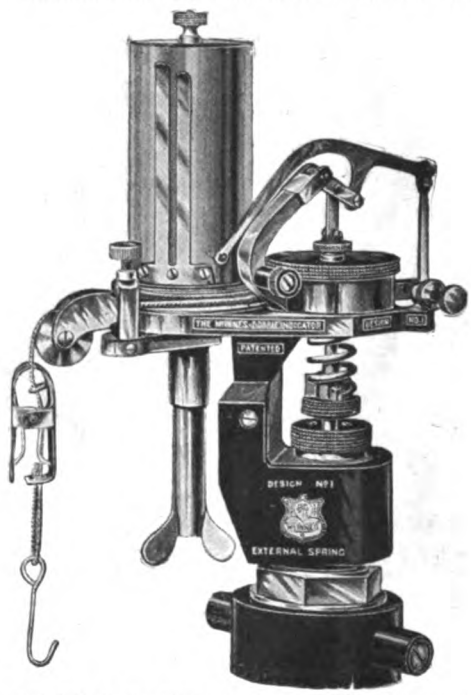
The McInnes-Dobbie Indicator.

In these indicators the pressure spring is placed above and outside the steam cylinder, a position in which it is kept comparatively cool when the indicator is under steam. For this reason the accuracy of the instrument cannot be impaired by the temperature of the steam. Not only this, but the spring can be very accurately calibrated under normal conditions and not at a high temperature, as is necessary when the spring is in contact with the steam.

The vulcanite cover caps above the piston chamber and the pressure spring unscrew, allowing the piston rod and parallel motion to be removed bodily from the instrument, for the purpose of changing the spring or cleaning the cylinder. These caps screw to the indicator frame and the lower cap acts as a guide to the piston rod, insuring perfect alinement and preventing escaping steam from passing upwards to the diagram.

The parallel motion is so constructed that the long lever carrying the marking point is not overhung, but is mounted so as to intersect the vertical axis of the pressure cylinder. It, therefore, receives a uniform and direct motion from the piston with a minimum liability of error, due to friction. In instruments having overhung arms a side strain, tending to bend the joint pins, is inevitable, and the friction thus set up results in a considerable drag on the linkage and in serious wear on the sockets and pins. The bearings of the several joints are broad and solid, specially hardened, and the motion is adapted to withstand severe strain without slackening at the joints, with corresponding play at the pencil point, which soon becomes a source of serious error.

The piston is of a patent type, made of steel, case-hardened, and it is claimed that it is unaffected by steam temperatures and does not expand and stick at high pressures. It is turned from a solid piece, and the central space affords accommodation for any grit present in the cylinder, removing it from the cylinder walls and preventing tearing and friction. The piston and all moving parts are of very light weight, in order to reduce to a minimum the inaccuracies due to inertia. The piston



travel is multiplied six times at the pencil point, consequently the piston itself has a very short travel. The drum spring is spiral, of light weight, and is fitted with brass ends. With this style of spring the tension of the drum can be conveniently altered to suit the speed of the engine, and in event of breakage can be readily replaced.

The indicators are manufactured by Dobbie McInnes, Ltd., Glasgow.

The Star Improved Indicator.

The Star improved indicator, manufactured by the Star Brass Manufacturing Company, Boston, Mass., consists of two main parts—the steam cylinder and the paper drum. The cylinder consists of an outer shell, which forms a part of the main body of the indicator, and an inner shell, in which the piston operates. The inner shell is removable and can easily be replaced when worn. The inner shell extends in one continuous piece of metal to the cap in order to obtain perfect alinement for the movement of the piston. At the lower end the inner shell is comparatively thin and it is surrounded by an annular space. This enables the working part of the cylinder to be free from any distortions that may result from difference in expansion in the inner and outer shells or from undue strain which may be brought to bear upon the body of the indicator. The upper part of the inner shell is surrounded by a channel and communicates with it by a number of holes drilled through the metal. From this channel openings through the body of the outer cylinder carry away the steam and vapor which, in the process of operation, blows by the piston.

The piston consists of a thin cylindrical shell with a transverse web across the center. It is made of tool steel hardened and ground, but the hub at the center of the web is of soft steel, to which the piston rod is screwed. By means of proper

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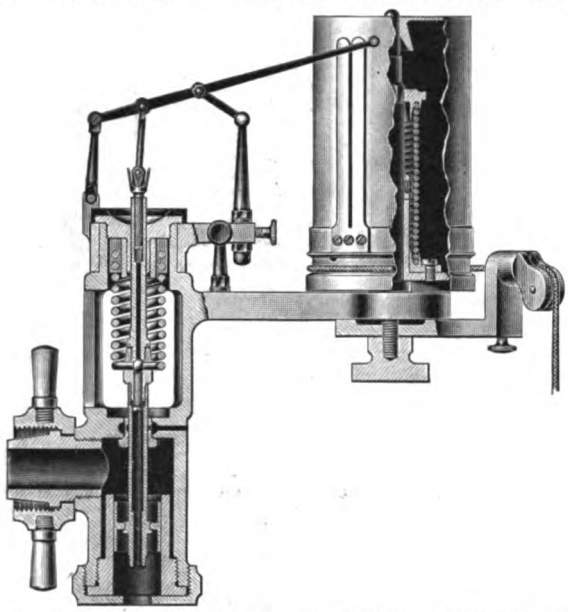
adjustments a flexible connection is provided between the piston and the spring. The upper end of the piston rod is hollow and threaded to receive a swivel head, by means of which any desired vertical adjustment of the position of the pencil can be secured without removing the cap from the cylinder.

The spring, which is one of the most important parts of the whole instrument, consists essentially of only two parts—the head upon which the spring is mounted, and the coil of wire which forms the spring itself. There is no metal at the lower end except a small sphere or ball through which the wire passes and to which it is attached. The ball forms the point



of attachment for the piston, furnishing a ball-and-socket joint. When it is considered that the momentum of the mass of metal in the spring has a decided effect upon the form of the diagram which the instrument produces, the advantage of this form of spring is evident, since the weight at the piston end has been reduced to a minimum.

The pencil movement consists of three parts, and since the degree of perfection with which the rectilinear motion is obtained in this movement is dependent solely upon the accuracy with which delicately-turned pins can be fitted to reamed holes, it is evident that the pencil movement can be made very accurate. The mechanism is so proportioned that the movement of the piston is multiplied at the pencil end six times. The design provides that the pencil does not deviate from a straight line in any sensible degree unless the extreme movement extends more than 1¼ inches from the central position.

Fulton Exhibit, Engineering Societies Building.

The Hudson-Fulton celebration at New York is essentially a recognition of the explorer and the engineer. To show the relation of the latter to the celebration, models of the *Clermont* and other early steamboats, through the courtesy of the Smithsonian Institution, are now on exhibition at the rooms of the American Society of Mechanical Engineers in the Engineering Societies' Building, 29 West Thirty-ninth street. The exhibit includes the *Clermont*, the *Phoenix*, built by John Stevens, and one of John Fitch's early types. Original drawings by Fulton, an oil portrait of Fulton, painted by himself, Fulton's dining table, oil portraits and bronze bust of John Ericsson, models of the *Monitor*, all owned by the society, and Ericsson's personal exhibit at the Centennial Exposition, are also exhibited. Through the courtesy of the Hamburg-American line a beautiful model of the *Deutchland* shows the highest type of the development of steam navigation.

The exhibit will be open to the public every week day from 9 A. M. to 5 P. M.

TECHNICAL PUBLICATIONS.

Reed's Polyglot Guide to the Marine Engine. Size, 11 by 8½ inches. Pages, 174. Numerous illustrations. Sunderland, 1909: Thomas Reed & Company, Ltd. Price, 6s. net.

This polyglot guide to the marine engine is intended to fill a long-felt want among marine engine builders, shipbuilders and seagoing engineers who are compelled to do business in several different countries involving dissimilar languages. The book is divided into eighteen sections, enumerating the various details of boilers, engines, engine-room stores and types of vessels. The parts are all numbered and many of them illustrated. The name of each part is given in English, French, German, Norwegian, Italian and Spanish. There are also alphabetical indexes for each of the languages and a table of monies, weights and measures.

Fighting Ships, 1909. Edited by Fred T. Jane. Size, 12 by 7½ inches. Pages, 492. Numerous illustrations. London, E. C., 1909: Sampson Low, Marston & Company, Ltd. Price. 21s. net.

Since extended reviews of the last two editions of this book have been published in this magazine, undoubtedly the majority of our readers are familiar with the general scope and arrangement of the work. No very striking change has been made in the general features of the present volume, which is the twelfth edition. As before, complete details of all the important warships in the world are given, together with illustrations showing the exterior of the ship and an outboard profile and deck plan, indicating the arrangement and distribution of guns and armor. Part II. contains a number of valuable and timely articles on various matters connected with warship design and construction, including the protection of battleships against submarine attack and the progress of warship engineering.

Probably the one thing which has occasioned more discussion and interest in warship construction during the last year than any other has been the comparative secrecy and mystery which have veiled the plans and development of the latest German battleships and battle cruisers. Considering the difficulty of obtaining information regarding these ships, remarkably complete details have been collected by the author, and, as he does not hesitate to state the source and probable accuracy of his information, the reader is able to form his own conclusions as to the probable value of the various reports and conjectures which are current.

In general, in the present edition, increased space is given to the plans and description of the smaller and less important vessels in various navies. Deck plans of destroyers and torpedo boats are given in nearly every case.

Steam Turbines. (The Power Hand-Book Series). By Hubert E. Collins. Size, 4½ by 6¾ inches. Pages, 186. Figures, 76. New York, 1909: Hill Publishing Company. Price, $1.00 net.

So many books have recently been published on steam turbines which take up the subject from a theoretical standpoint that many practical operating engineers will welcome this little book, which is simply a practical description of a few of the leading types of steam turbines together with some practical notes which have been gathered from the experience of successful engineers in operating steam turbines. The volume includes chapters describing the Curtis, the Allis-Chalmers and the Westinghouse-Parsons turbines, the method of setting the valves of the Curtis turbine, the proper method of testing a steam turbine, auxiliaries for steam turbines and trouble with steam turbine auxiliaries.

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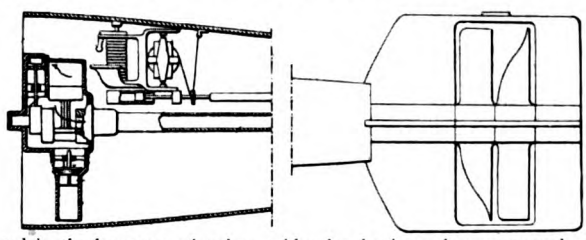
SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

917,449. SELF-PROPELLED TORPEDO. ALBERT EDWARD JONES, OF FIUME, AUSTRIA-HUNGARY, ASSIGNOR TO WHITEHEAD & COMPANY, OF FIUME, AUSTRIA-HUNGARY, A CORPORATION.

Claim 1.—In a self-propelled torpedo, the combination of a gyroscope placed above the longitudinal axis of the torpedo, a buoyancy chamber into which the servo-motor of the gyroscope discharges, a sinking valve



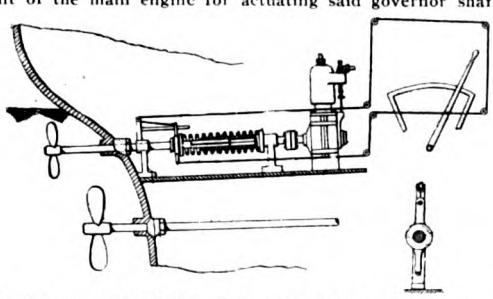
placed in the buoyancy chamber, said valve having a box connected with the exhaust chamber of the engine and a socket opening in close proximity to the bottom of the buoyancy chamber. Four claims.

920,286. FLOATING DRYDOCK. WILLIAM THOMAS DONNELLY, BROOKLYN, N. Y.

Claim 1.—In a floating drydock having separate water compartments, a centrifugal pump in each of said water compartments for the admission and exhaust of water to and from said compartments, power connections for each pump extending from the upper portion of the dock, each of said pumps establishing communication only with the compartment in which it is located and the exterior of the dock and having a combined inlet and outlet to the exterior of the dock, a combined inlet to and outlet from the compartment in which it is located and a single valve for each of said pumps for controlling the passage of water therethrough. Two claims.

921,714. MARINE-ENGINE GOVERNOR. LEWIN D. KISSACK, OF CLOVERDALE, CAL.

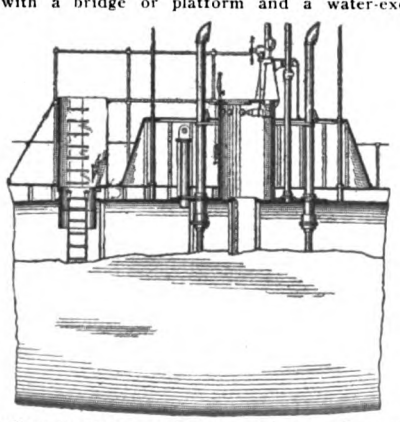
Claim. 1.—The combination with a marine vessel, its propeller shaft, and engine throttle operating means, of a governor shaft, means independent of the main engine for actuating said governor shaft, and a



connection between said shaft and the throttle-operating means arranged to move the operating means toward the closed position upon the emerging of the rear end of the propeller shaft from the water. Two claims.

922,056. SUBMARINE BOAT. LAWRENCE Y. SPEAR, OF QUINCY, MASS.

Claim. 1.—A submarine or submergible boat having a deck hatch in combination with a bridge or platform and a water-excluding hatch



trunk surrounding the hatch and cover and extending up to the platform. Three claims.

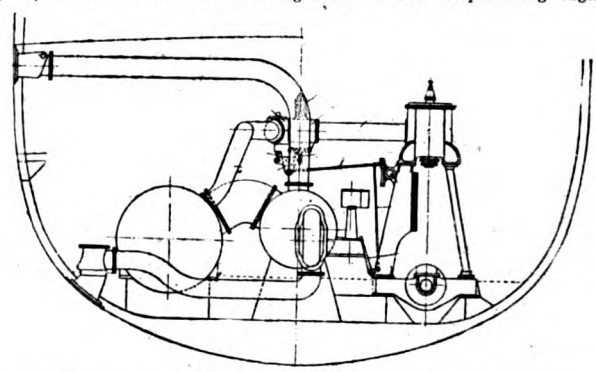
926,475. ACETYLENE-GAS BUOY. ROBERT M. DIXON, OF EAST ORANGE, N. J., ASSIGNOR TO THE SAFETY CAR HEATING & LIGHTING COMPANY, A CORPORATION OF NEW JERSEY.

Claim 4.- In a device of the nature described, in combination, a float comprising a chamber for the storage of gas, said chamber being at all times free from water, a submerged receptacle carried by said float and adapted to contain calcium carbid, a connection between said receptacle and said chamber for allowing the generated gas to pass from the former to the latter, a water inlet for said receptacle communicating with the water within which it is submerged, and automatically operating means for controlling the flow of water through said inlet. Ten claims.

British patents compiled by Edwards & Co., chartered patent agents and engineers, Chancery Lane Station Chambers, London, W. C.

12,538. SHIPS' PROPELLING MACHINERY. R. R. BEVIS, BIRKENHEAD, AND J. H. GIBSON, LISCARD.

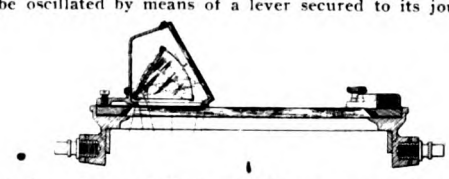
Claim.—According to this invention, the control of mixed machinery, i. e., rotary and reciprocating engines, for propelling, is effected by a single mechanism. In the example given, this is done through the agency of a link on the reversing shaft of the reciprocating engine.



The rod for actuating the reversing valves of the turbine is connected to a block sliding in this link, and by placing the block at one end of the link the rotary engines may be caused to rotate in the same direction as the reciprocating engines, or, if the block be at the other end of the link, in the opposite direction.

18,510. ATTACHMENT FOR MARINERS' COMPASSES. L. W. P. CHETWYND, KINGSTON-ON-THAMES; KELVIN & WHITE, LTD., GLASGOW, AND F. W. CLARK, GLASGOW.

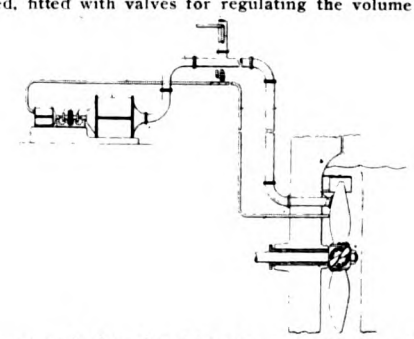
Claim.—Steering or bearing prisms are enclosed in a hood secured to the clamping ring or other part of the compass bowl. This hood is open at bottom and the prisms is mounted within upon a pivot so that it may be oscillated by means of a lever secured to its journal and ex-



tending to the exterior of the hood. Here, a segmental strip notched or indented retains the lever in any desired position. The front of the hood is covered with a glass pane, and the open bottom is surrounded by a groove for containing packing, which forms a joint with the glass of the bowl, so that the hood is inaccessible to moisture or dust.

19,047. STARTING AND REGULATING THE SPEED OF MARINE ENGINES. F. H. TANNER, HILLSIDE, STAPLE HILL ROAD, FISHPONDS, BRISTOL.

Claim.—In order to avoid difficulties in starting engines which are directly connected with the propeller, variable speed turbine compressors are employed, fitted with valves for regulating the volume and pressure



of the air or gas to be introduced into the area of action of the propeller through pipes. A shield prevents the air from escaping too readily to the surface of the water.

15,007. SIGNALING OVERHEATING OF BEARINGS. W. VOLKMER, GLEIWITZ, GERMANY.

Claim.—A block of fusible metal rests against the bearing and is carried at the top of a lever pivoted on pins. When the metal is melted by the hot bearing, the lever is rocked on its pivot by a spring, so as to release a toothed wheel and allow the clockwork to ring a bell. When the block is under the bearing, the rod must slide lengthwise, so the eccentric piece is turned to fill the slot and allow the lever to slide on the pins. The spring is then placed between caps, so that when the block melts, the spring raises the rod to release the bell mechanism.

19,446. ARMOR PLATES. M. R. RIDOLFI, FLORENCE, ITALY.

Claim.—In order to break the point of the projectile and so alter its angle and lessen its penetrative power, the case-hardened plate proper is provided with an outer hardened and toughened steel plate backed by wood. In addition to the above impediments, the projectile is further hampered by a plate, which, when pierced, forms a tough restraining collar around the nose of the projectile. A wood cushion is provided next the side of the ship.

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